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Vol. 44 No. 8 AUGUST 1954 PRICE: 3s. 6d.

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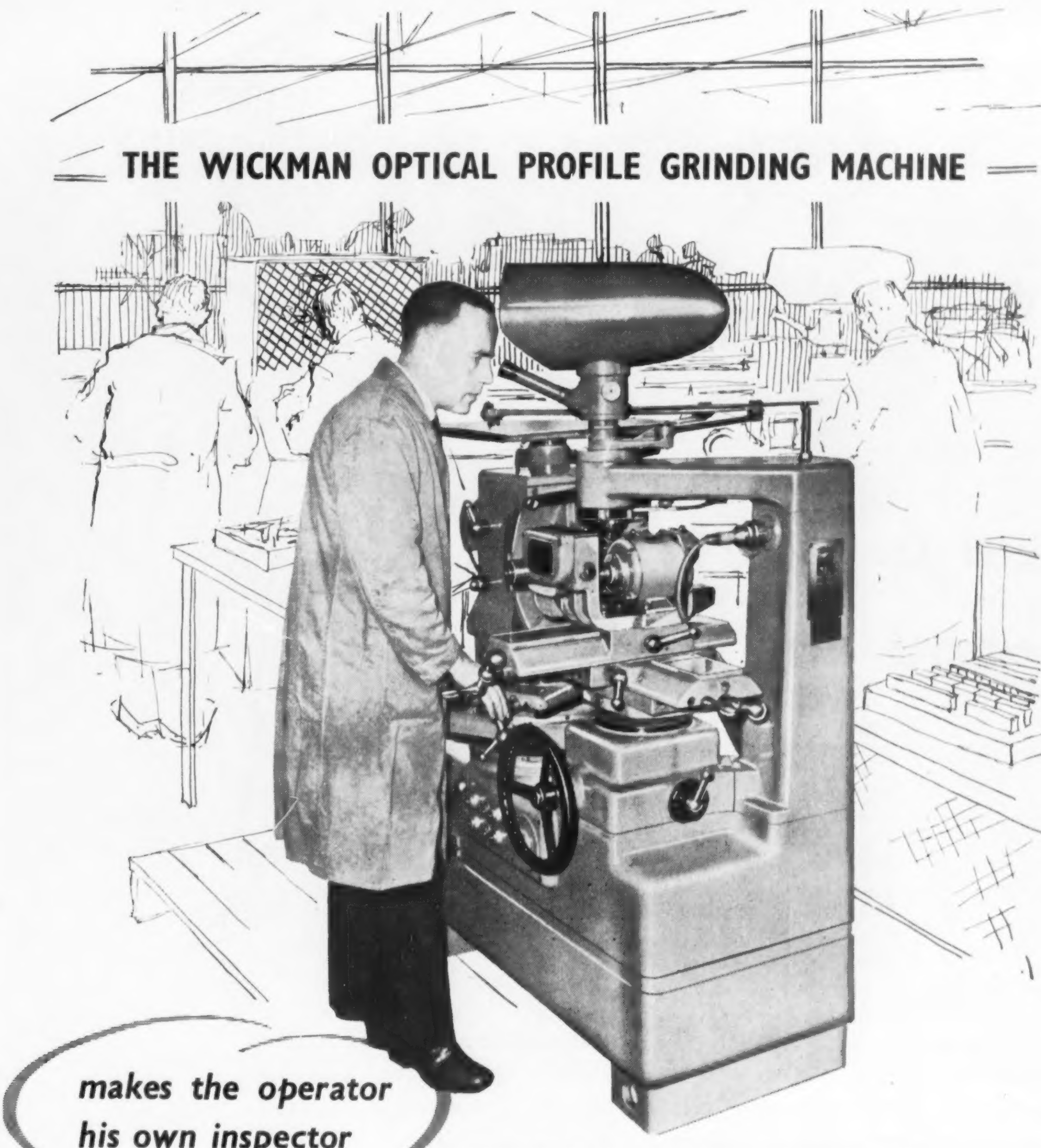

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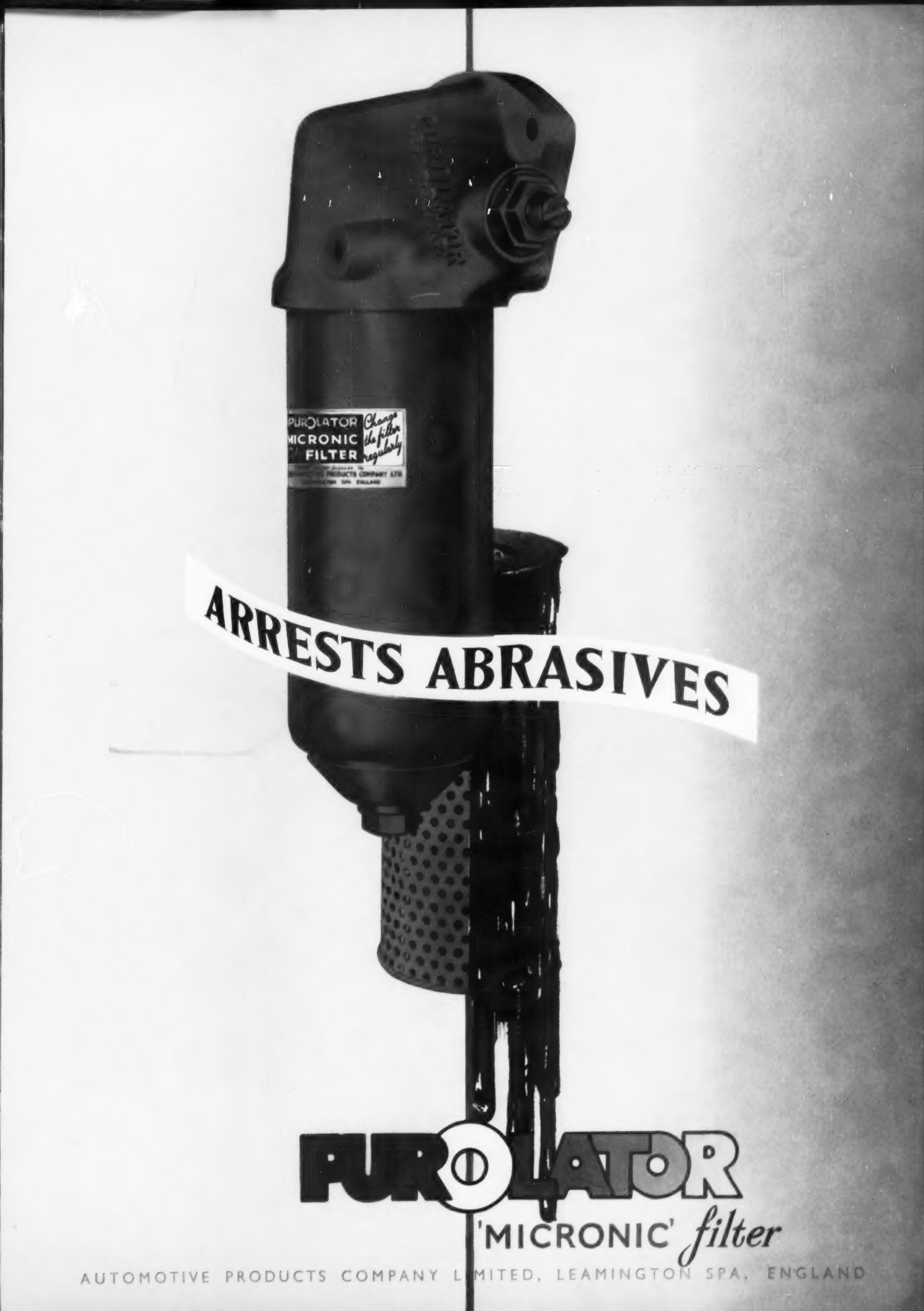
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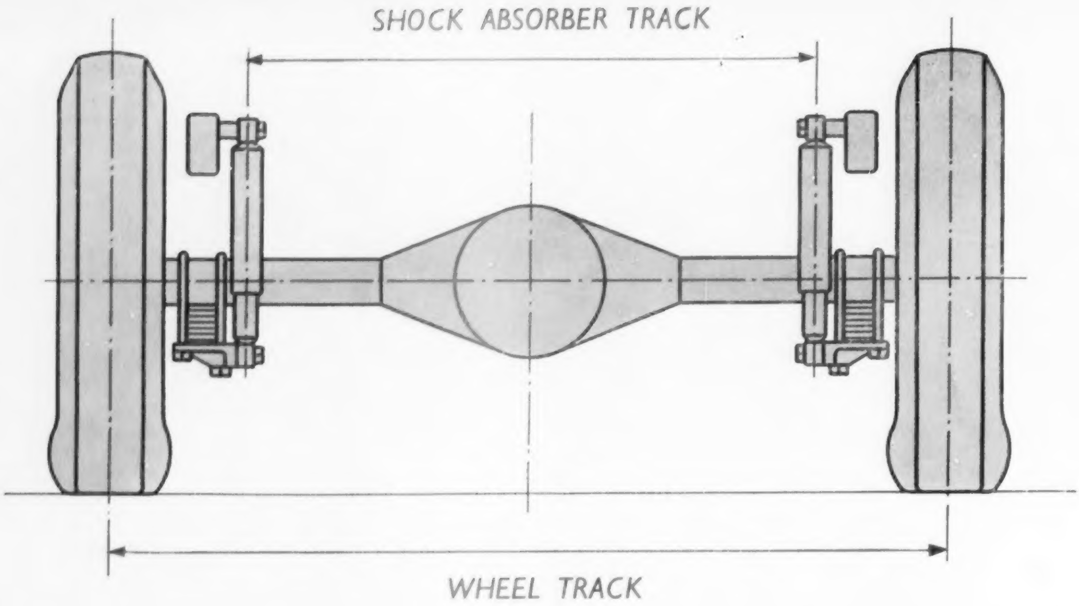

SUBTLETIES OF STEERING

The Transient Stage: 4. Shock Absorbers

Independent front suspension is now fairly general, and even with a fairly forward centre of gravity there is, anti-roll rod apart, probably not a great deal of difference between the anti-roll stiffness of front and rear suspensions, so long as we retain the rear axle with leaf springs. Since the roll centre at the front of the car is seldom far above the ground, whereas that at the back is probably well above the ground, there is therefore likely to be a greater weight transference at the rear while cornering in steady conditions. This is the reason for the desirability of the anti-roll rod at the front, since there it introduces extra weight transference at that end of the vehicle and hence promotes the desired under-steering effect.

With independent front suspension the shock absorbers have the same effect at the wheel in roll and ride deflections.

With an axle this is not the case; the effect on roll may be as little as half that on ride, and it is difficult to make it more than $\frac{2}{3}$; this is because the effect is diminished in the square of the ratio of the shock absorber track to the wheel track, and the shock absorber track is unlikely to be more than 0·8 of the wheel track.

So long as the roll angle is changing, the shock absorbers are influencing the weight transfer at the two ends of the vehicle and are therefore affecting the handling.

So long also as we have a rear axle it requires a considerably greater damper load setting at the wheel than is needed at the front with independent front suspension—maybe three or four times as much. Do not be misled by the occasional car with independent front suspension which has the same damper setting front and rear; the leverage ratio in front brings down the damper settings at the wheel in the appropriate degree. If anything, therefore, the load transference due to change of roll angle is likely to be greater at the rear than at the front of the car. This means that while entering a bend the 'roll stiffness' weight transference (including shock absorber effect) is almost certainly greater at the rear than at the front, unless we have a front anti-roll rod, when it is likely to be more in front than at the rear. Two factors, however, tend to avoid any over-steer effect from extra roll stiffness at the rear, if it exists, as roll commences. The first we have already seen and it is the necessity of extra sideways load at the front to give the car the necessary rotational acceleration to enter the bend. The second is that the sideways load at the rear at the commencement of a turn is very small, also as we have already seen, and increases only gradually, and the factor which normally gives up to about half the rear end weight transference in steady cornering, contributes far less than this. We see, in fact, that there exist even more reasons for under-steer effect in entering a bend than we mentioned in the essay on this particular subject.

Let us now see how the shock absorbers affect the behaviour when leaving a bend. The first effect of straightening out, once this has begun to happen, is to reduce the total sideways load, and hence the roll angle. The shock absorbers resist this reduction of roll angle. Since their effect in doing so (with a rear axle) is probably more at the rear than at the front, the total weight transference at the rear will be reduced more in proportion than at the front. The under-steer tendency from this is in most cases less than the over-steer effect of the extra sideways load at the rear referred to in the essay on entering and leaving bends, and while helpful is inadequately so for the circumstances. Leaving a bend is still a more hazardous and uncertain operation than entering it, though the relative hazardousness will depend on the degree of under- or over-steer.

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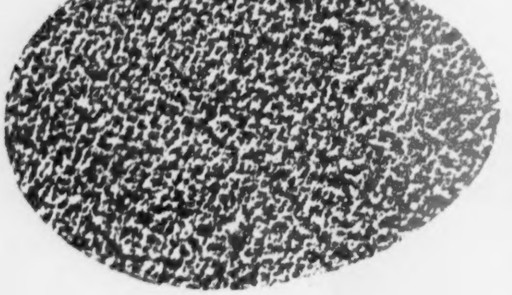

or the Egg ?

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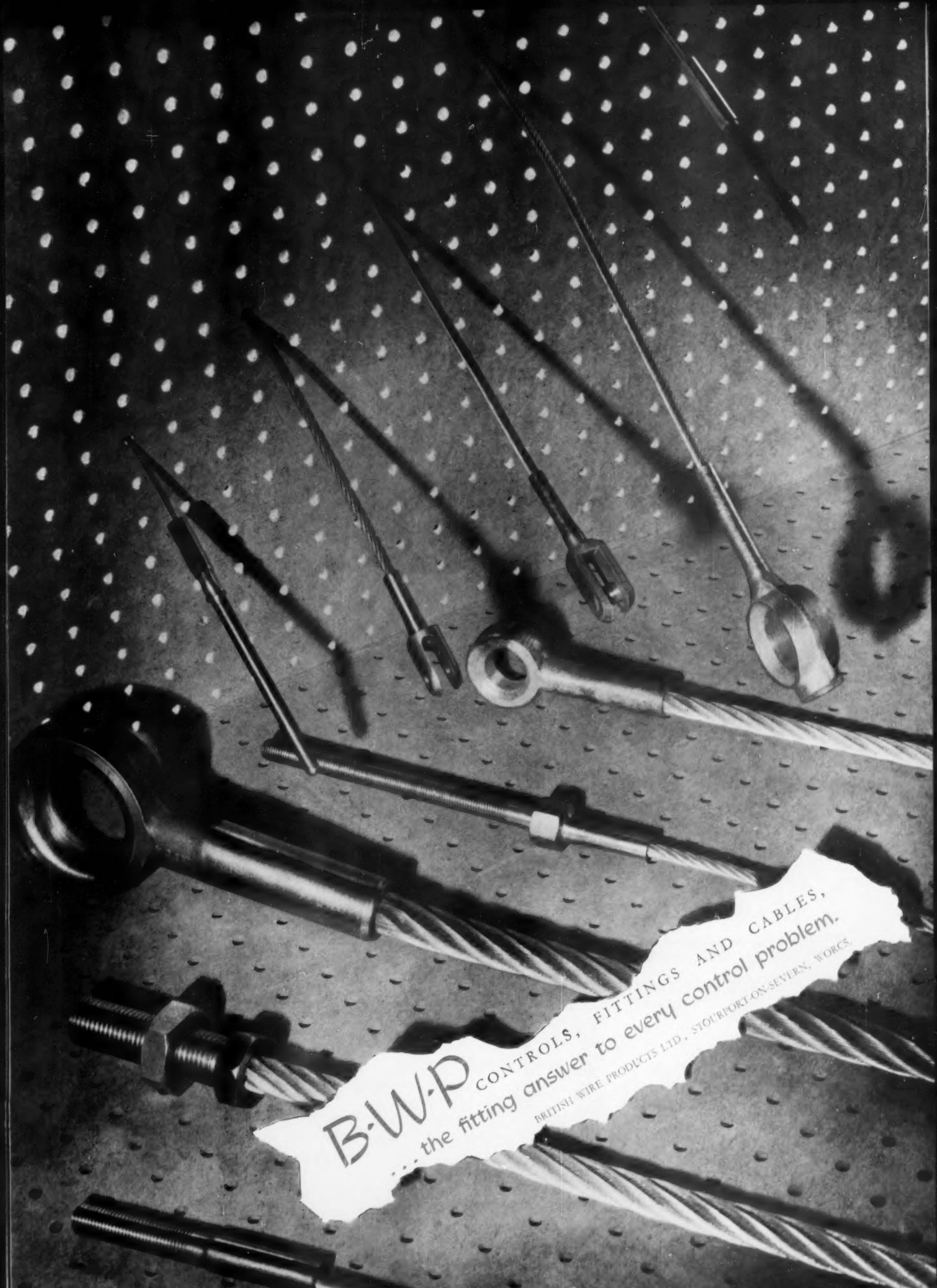
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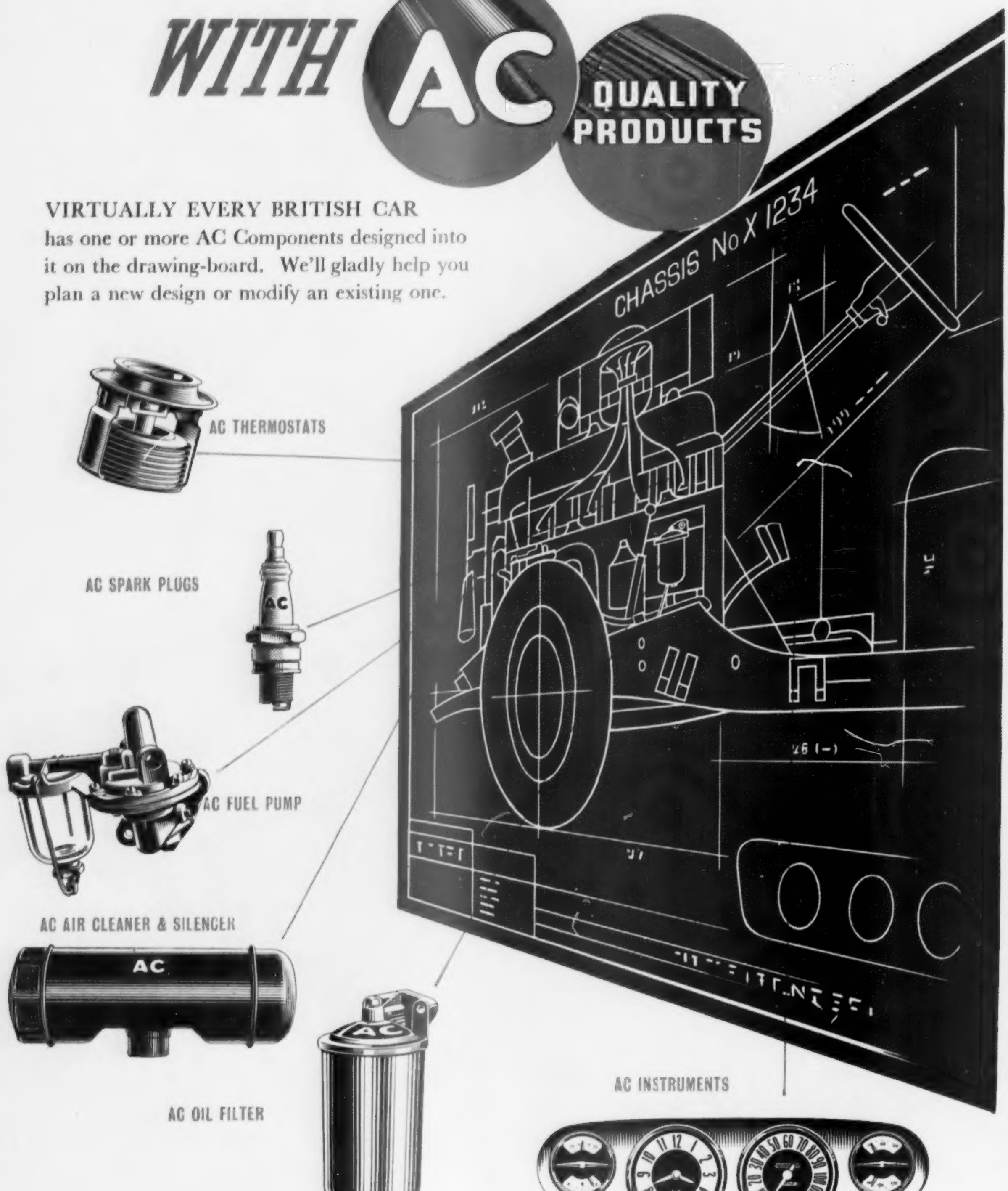

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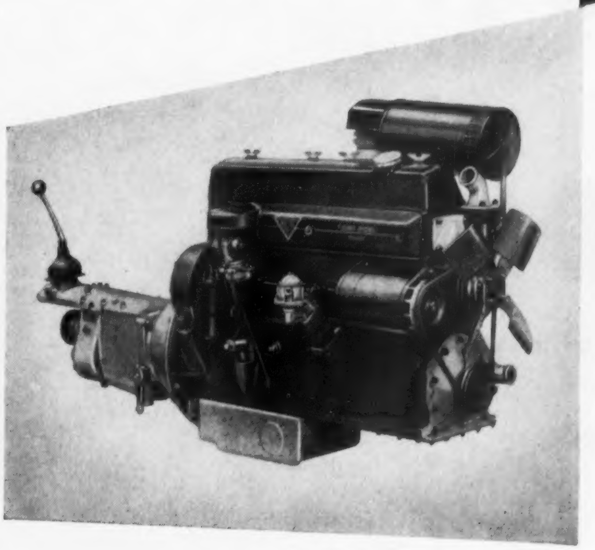

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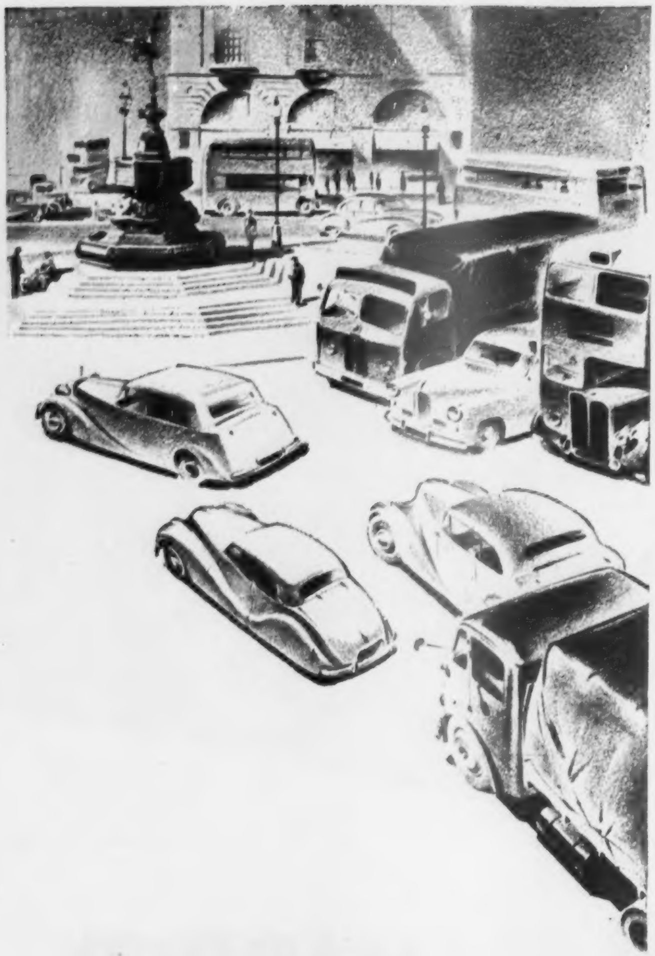

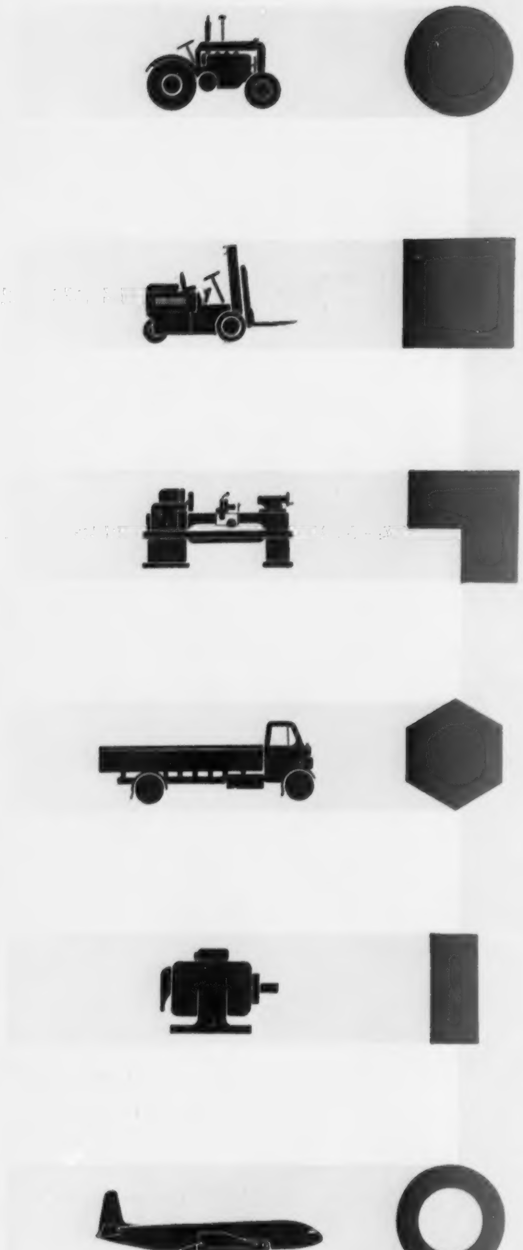

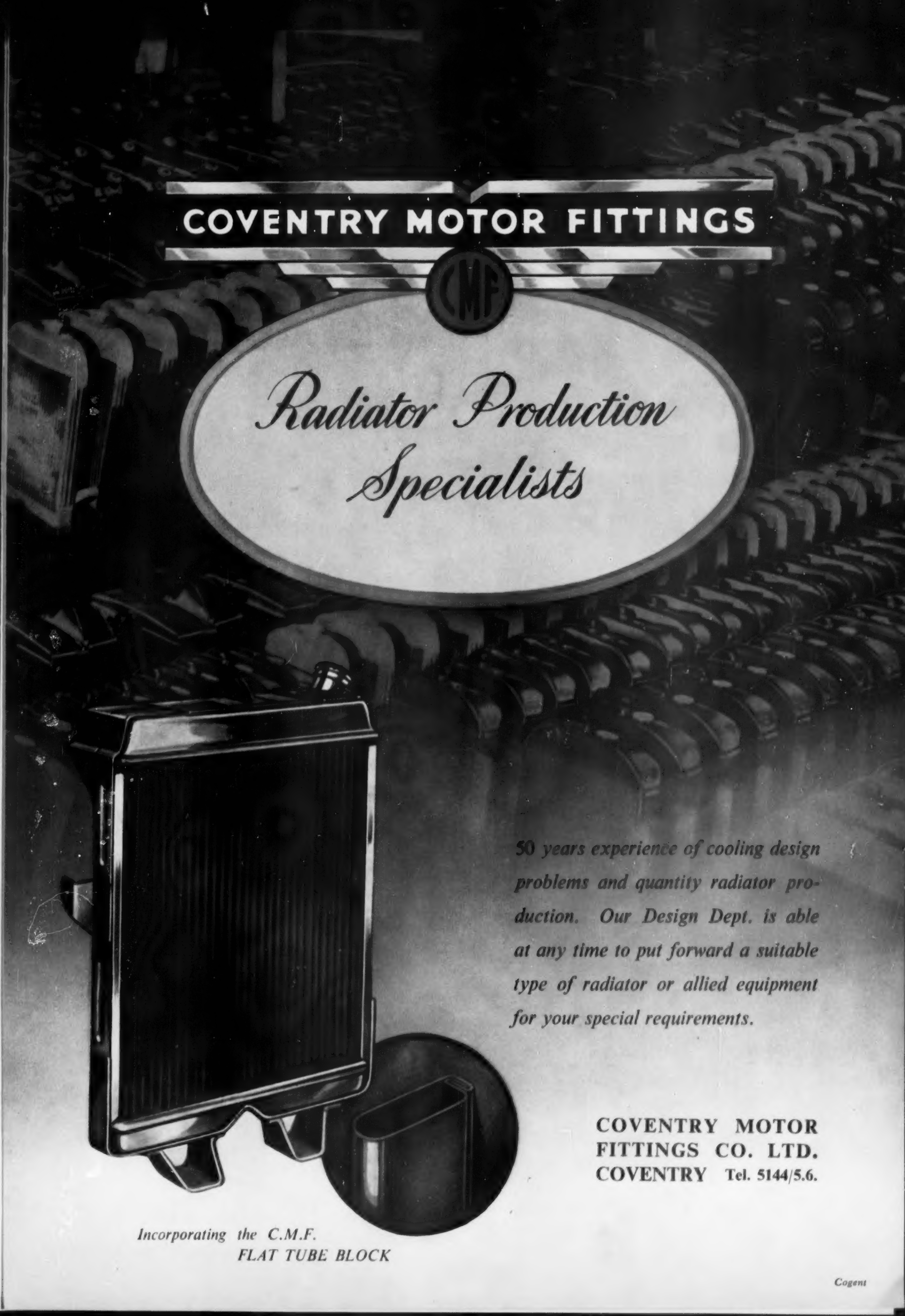
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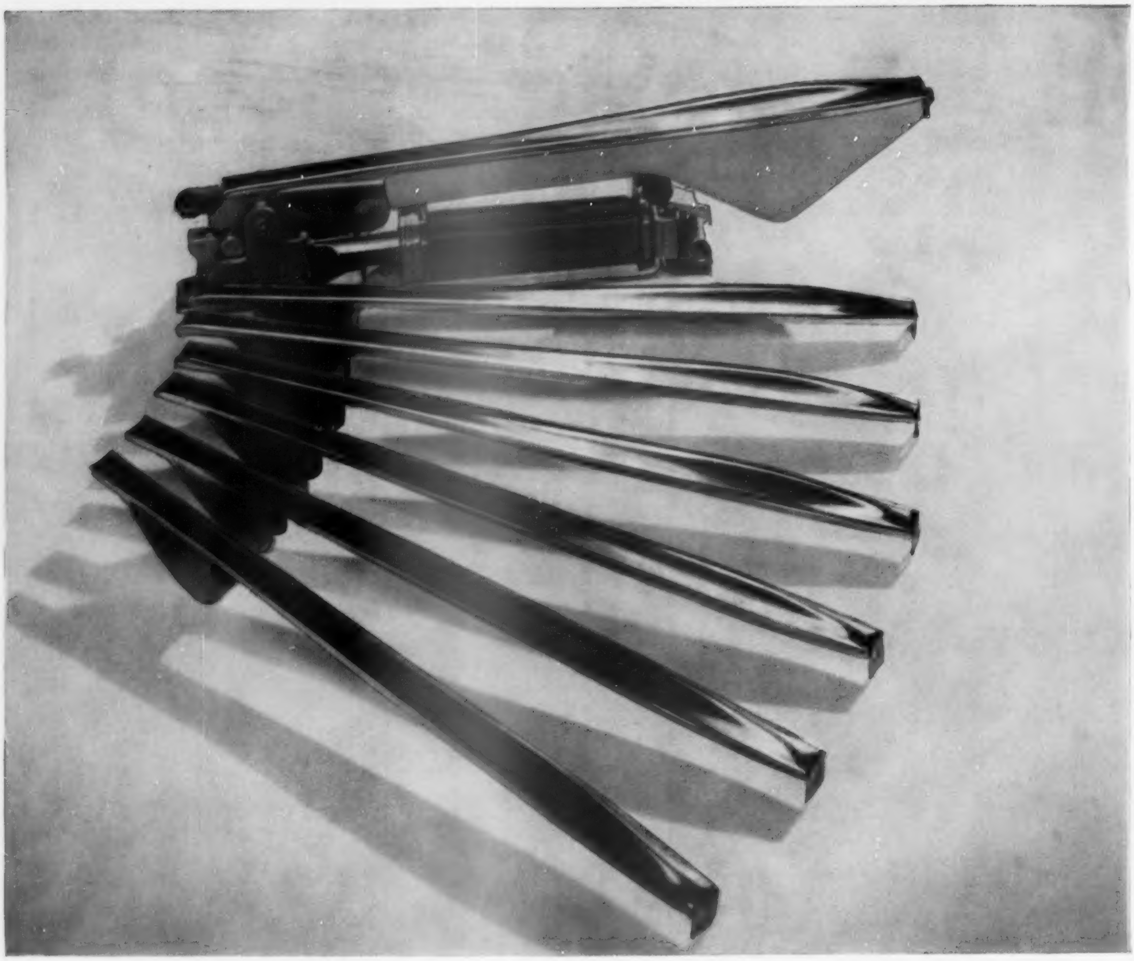

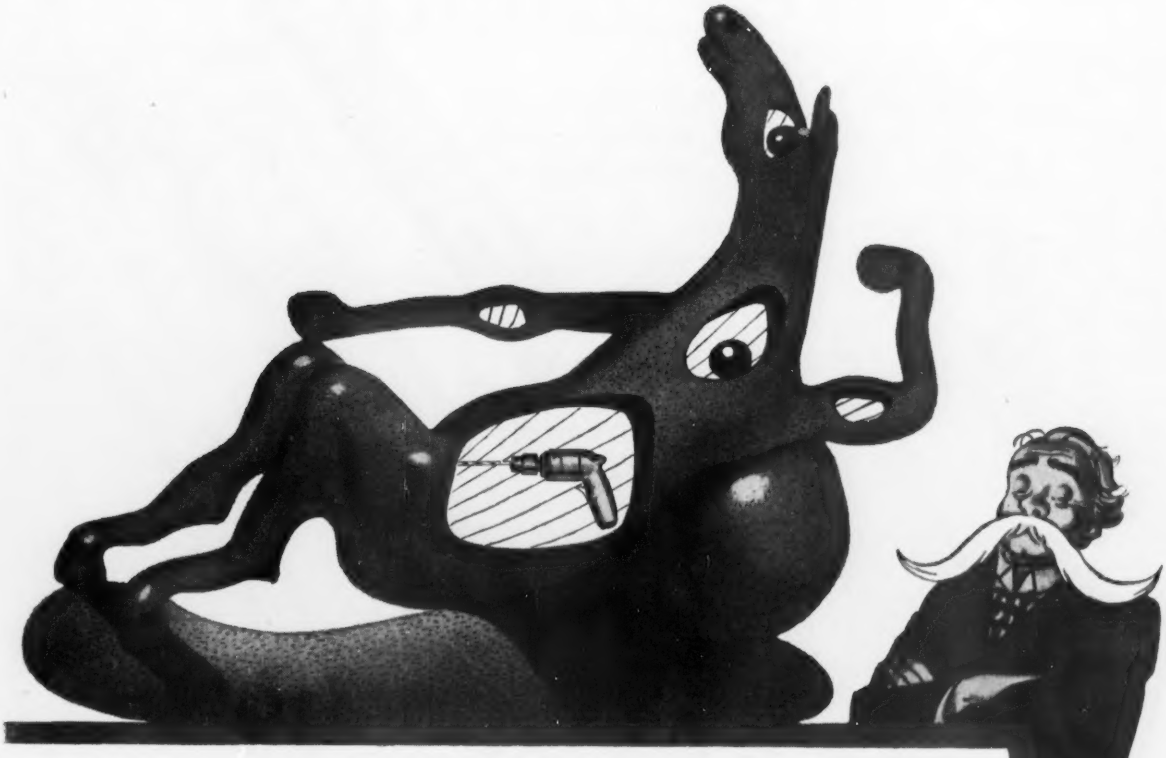

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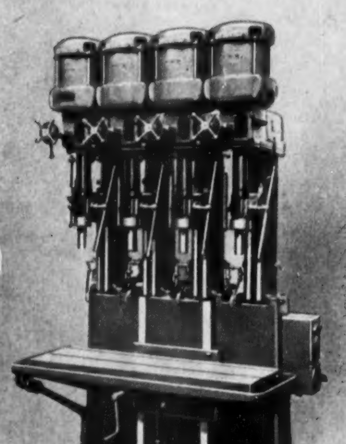

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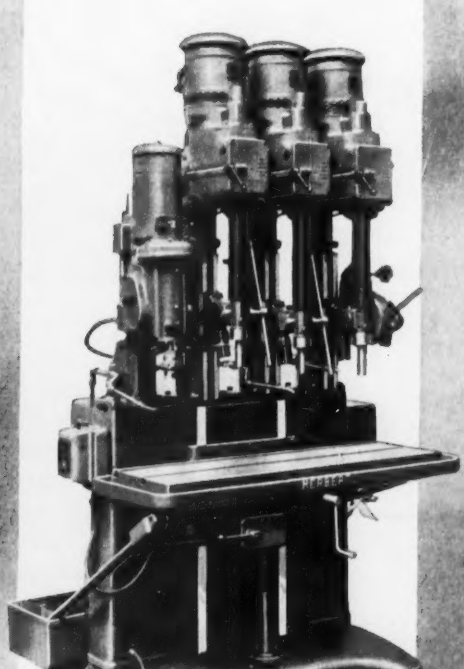

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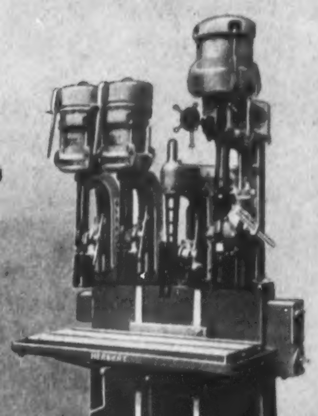

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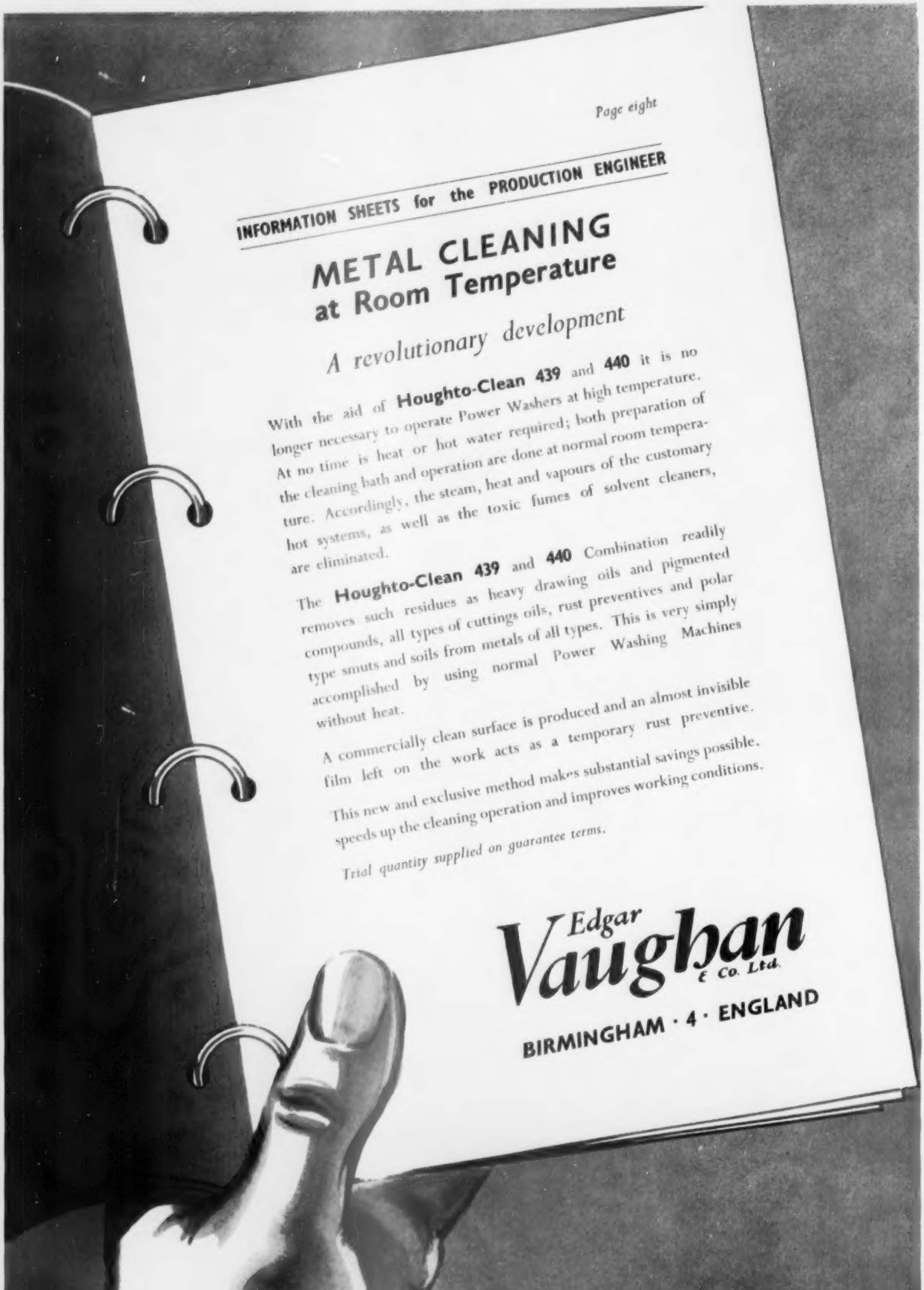
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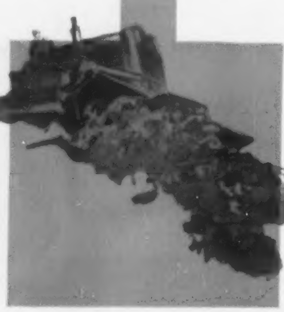

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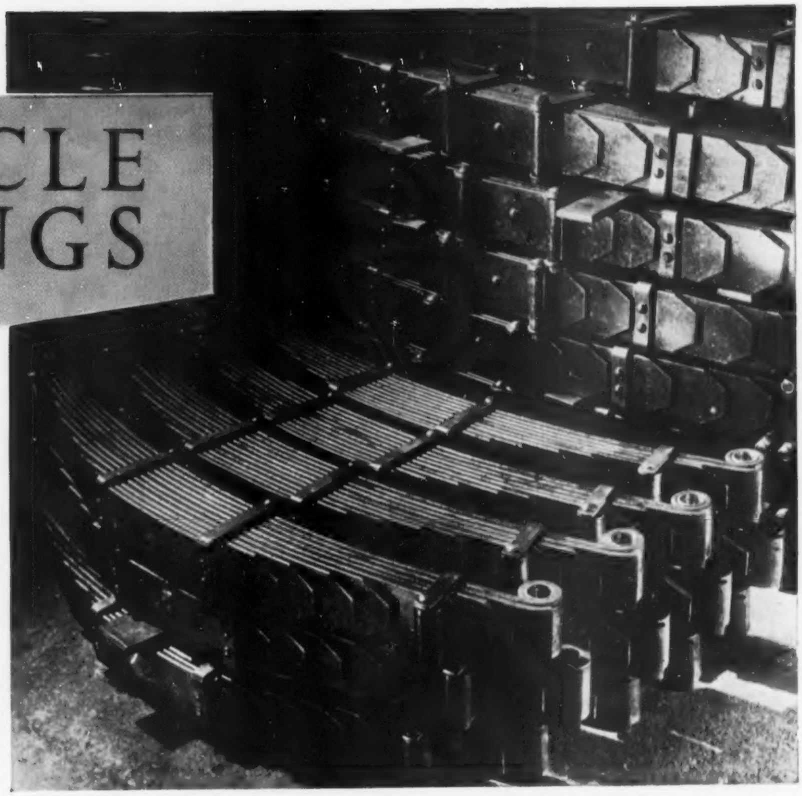

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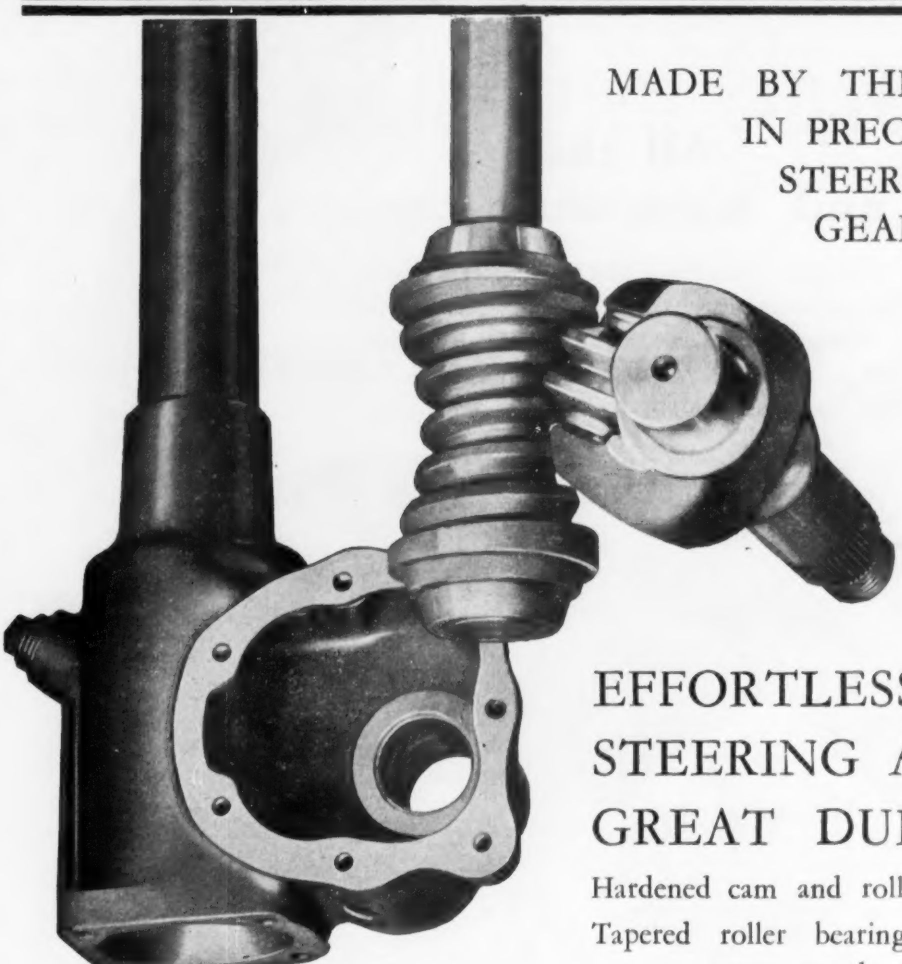

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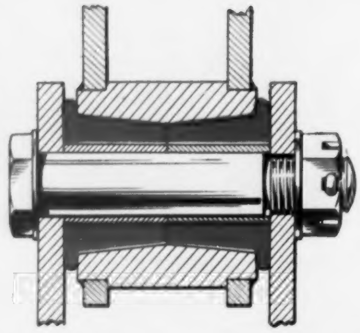

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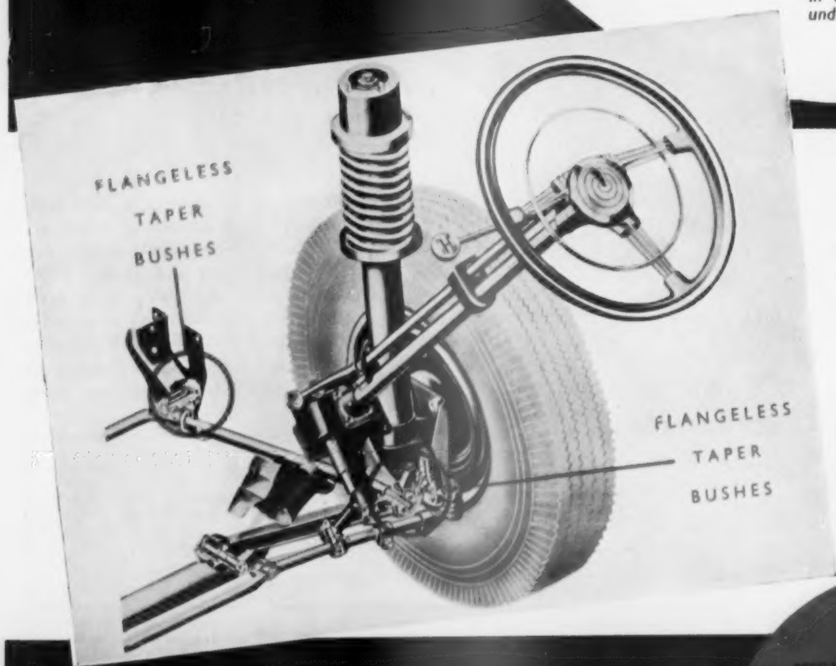

Left: Front Suspension of the "Consul" and "Zephyr Six".

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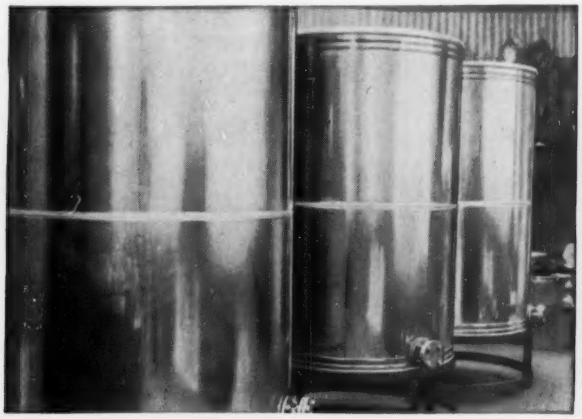

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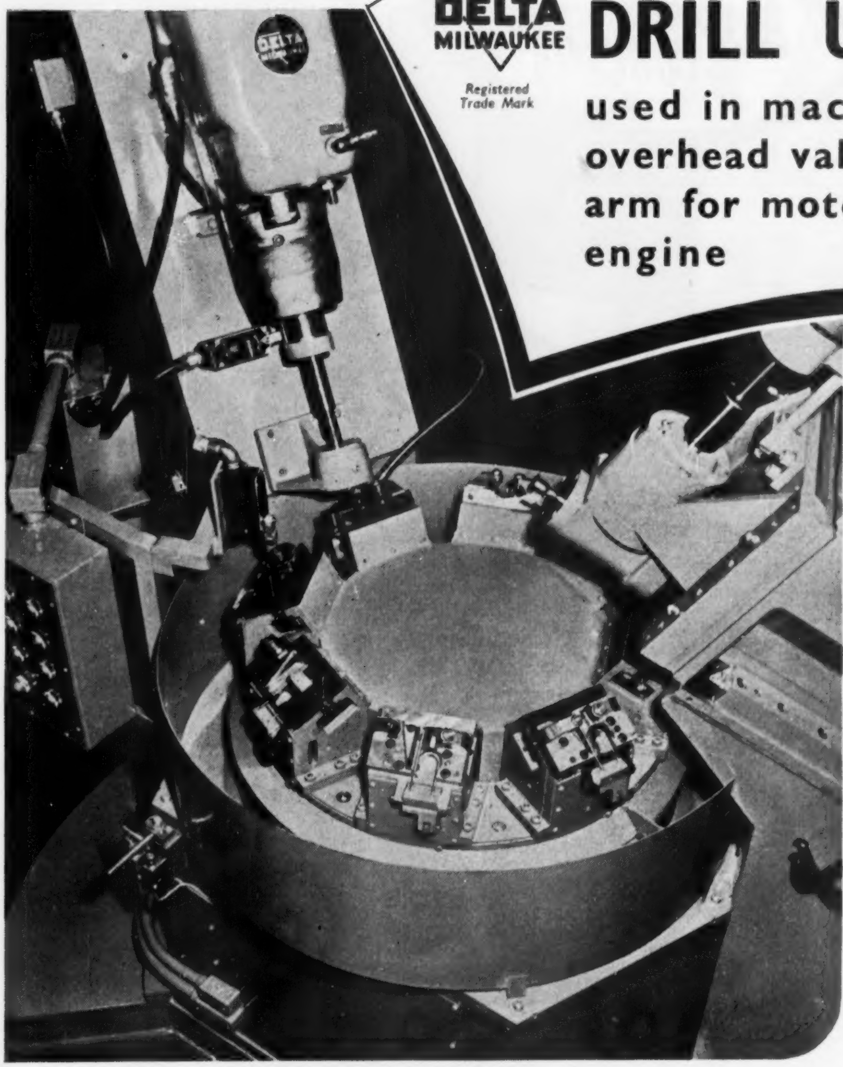

Illustration shows a special machine built around two Delta Milwaukee Model 19-400 Drill Units using a V-belt drive.

The part machined is the overhead valve rocker arm for a motor valve engine. The machine is constructed around a Geneva motion indexing table and it is interesting to note the boring operation carried on by the drill unit to the left. Here the spindle has been well supported by an external bearing to relieve the drill unit of the side loads encountered in the boring operation allowing the drill unit to serve as a prime mover only. The second working station involves a right angle milling slotting operation and it will be noted that the right angle milling head is well supported inways so again the drill unit is relieved of the side loads encountered by the offset machining operation.

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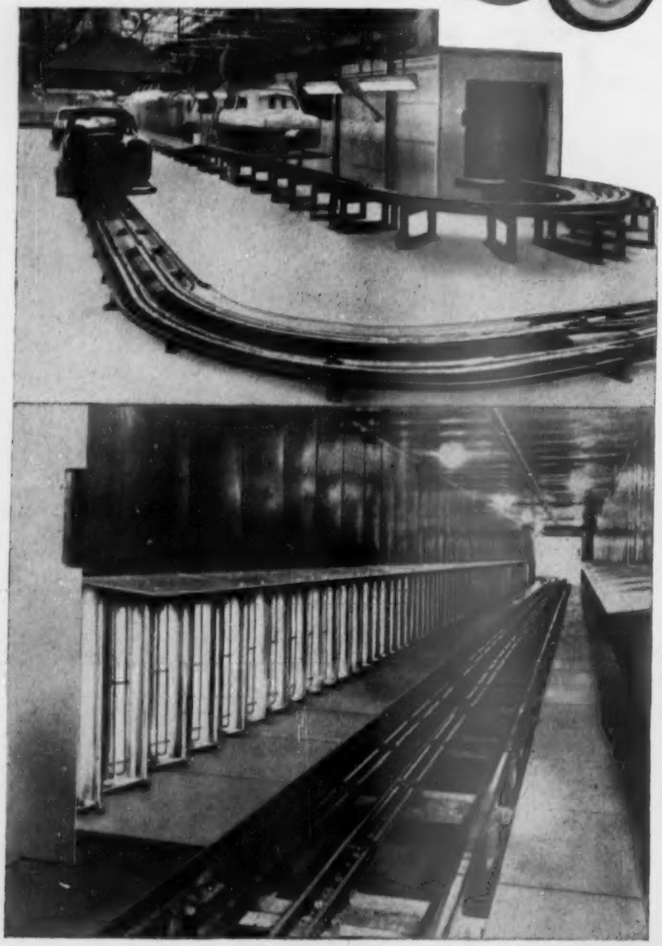

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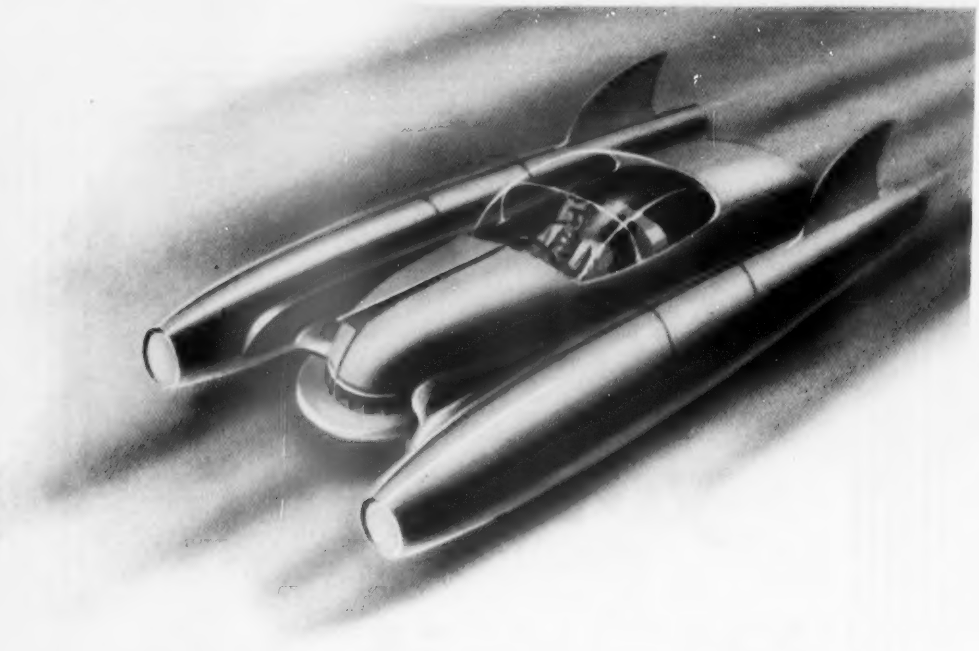

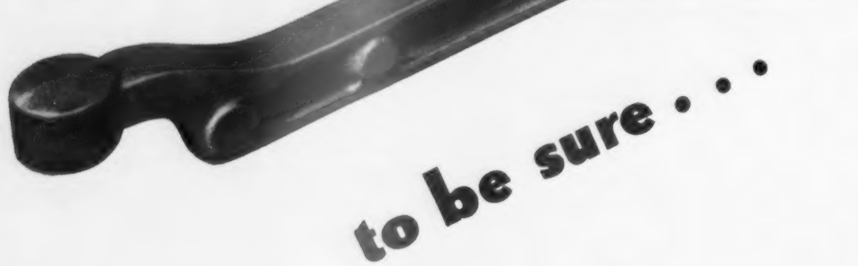

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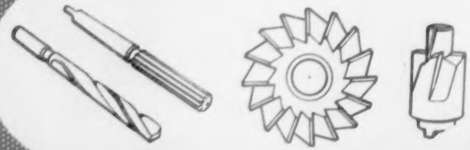

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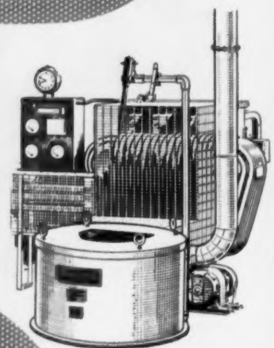

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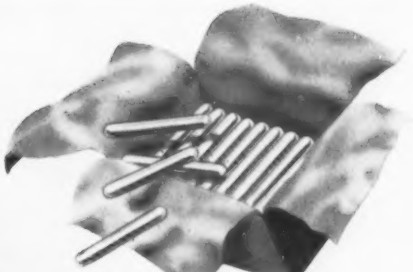

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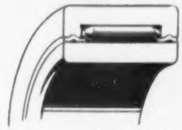

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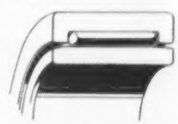

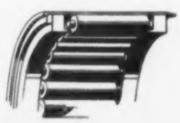

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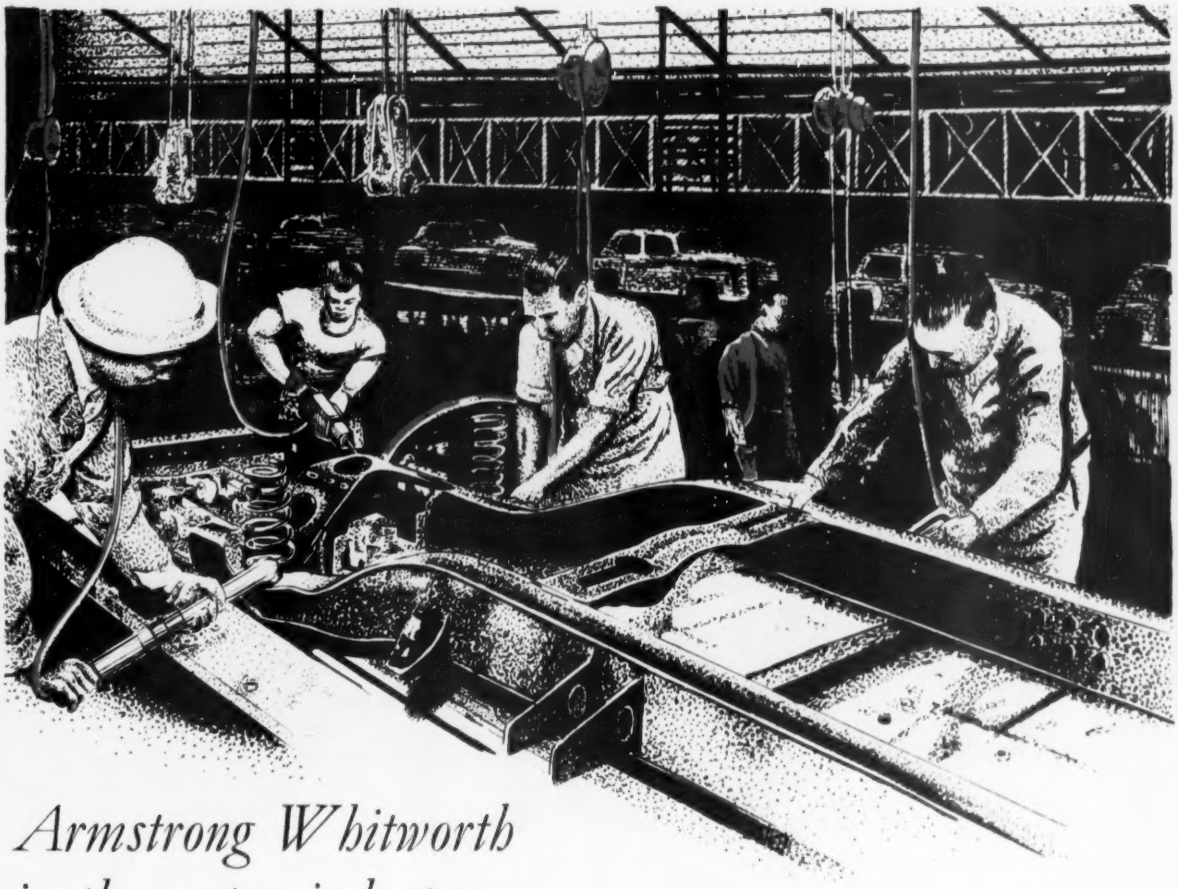

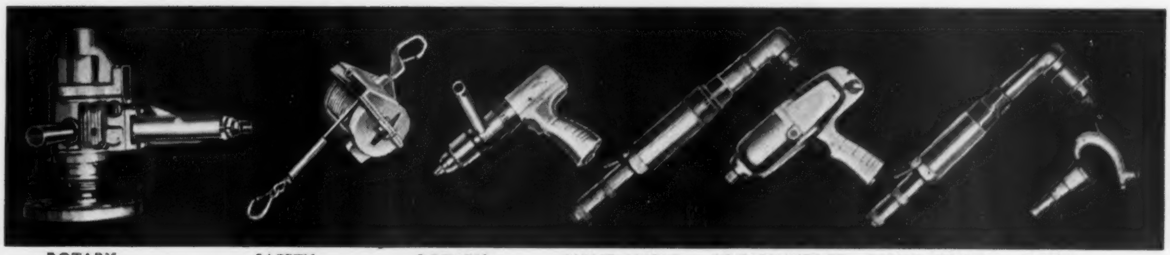

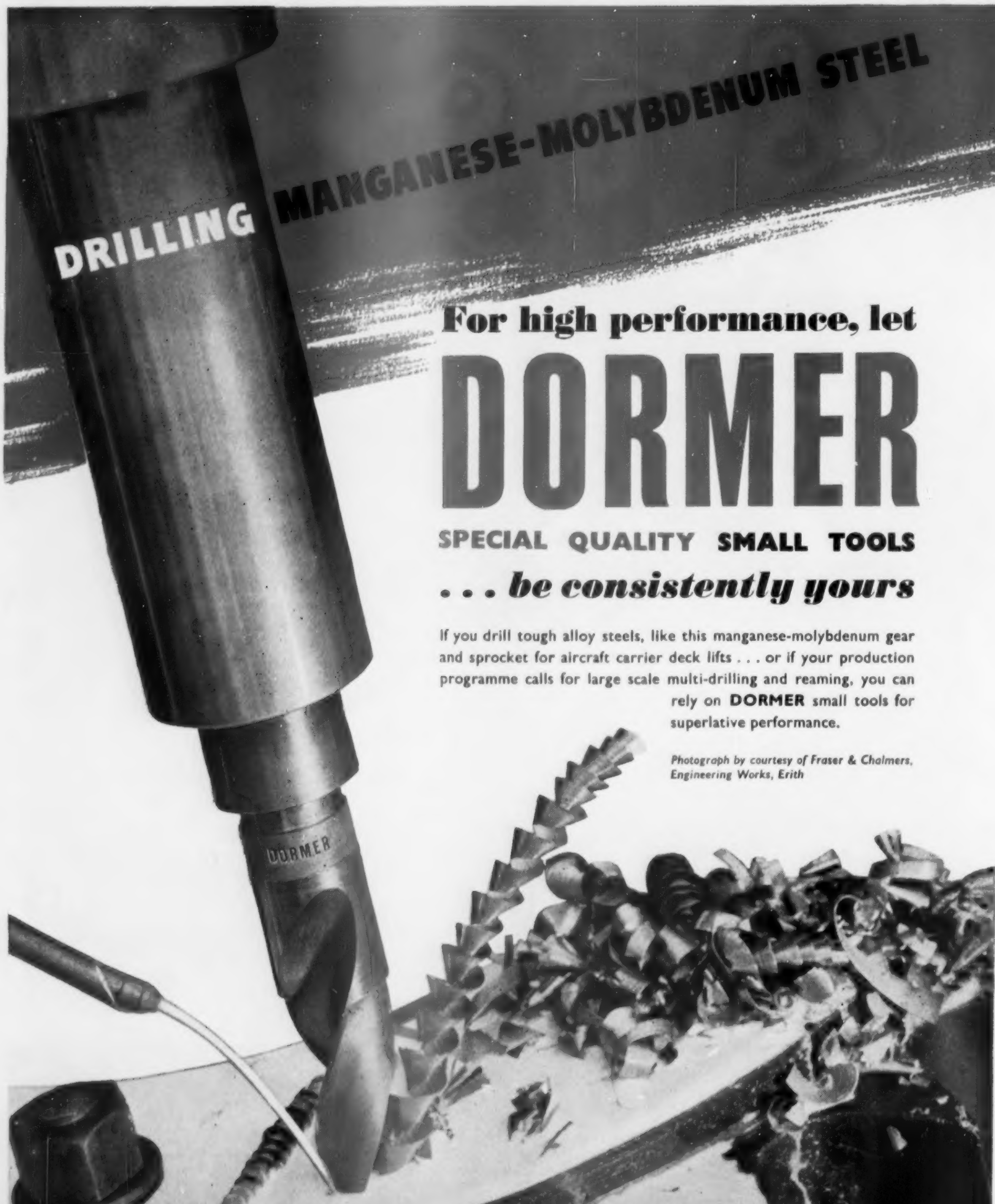
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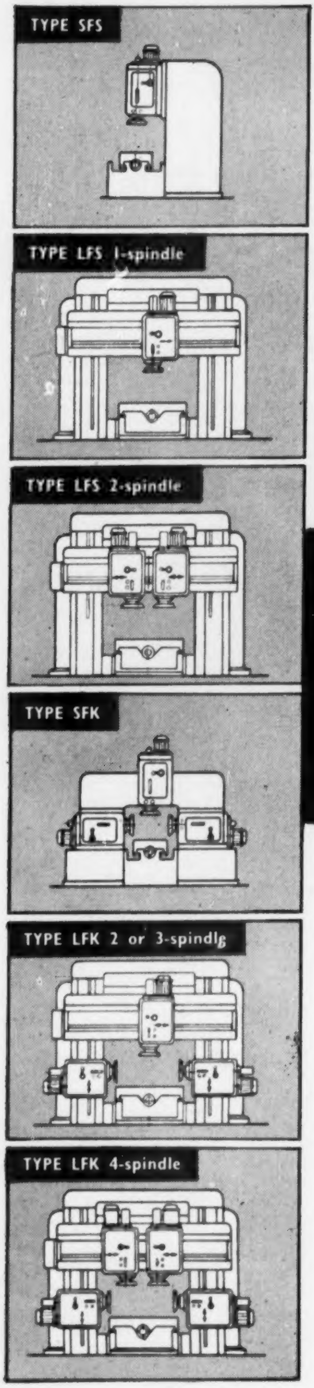

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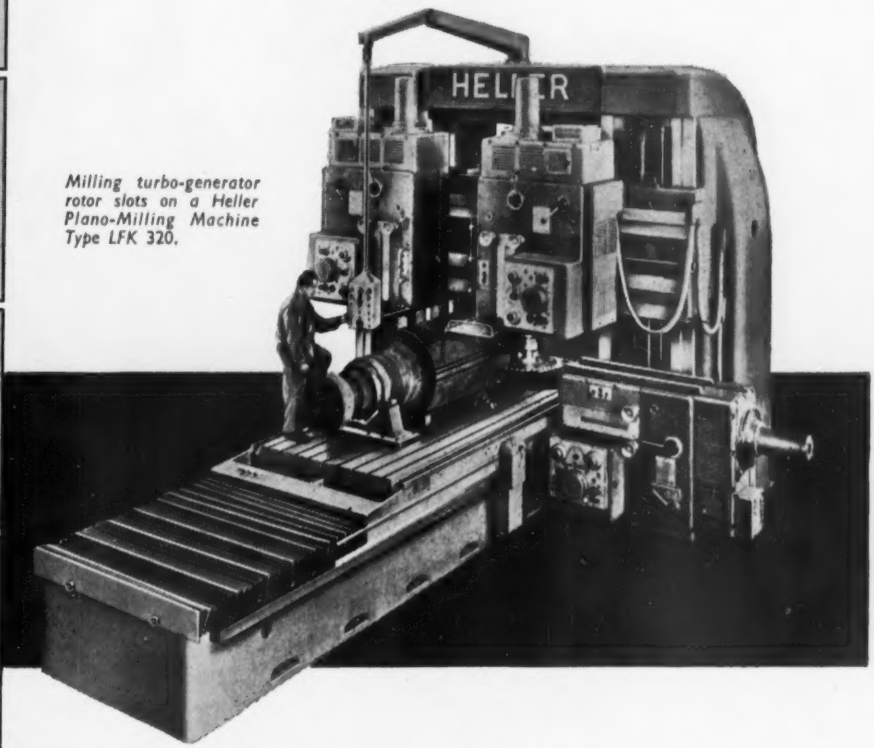

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Applied Research

TWO questions of great importance to-day are: Is this country devoting sufficient of its resources to applied research and development, and is the best use being made of the resources so employed? To the first, we feel that the answer is, No, although it is difficult to produce definite evidence; to the second, a more qualified No is probably justified. In fundamental research this country has an unsurpassed record; but, except in certain fields, it lags behind the United States of America. It also lagged behind Germany in pre-war years, and there is evidence that in some branches Western Germany is now abreast if not ahead of us. Yet, as Sir Henry Tizard has pointed out, Britain's main effort should now be in the field of applied research since our need is to develop new and better products for export and to find new and better production methods.

In view of the remarks in the preceding paragraph it may seem paradoxical to claim that pure and applied science in this country have a reputation and recognition greater than can be found in any other country. For example, one of the highest honours to be gained in this country is to become a Fellow of the Royal Society, and in no other country is there an institution similar to the Parliamentary and Scientific Committee on which Members of Parliament meet and discuss scientific, technical and related problems with men actively engaged in research.

Official grants

Perhaps more important than what may be called prestige recognition is the fact that the grants from public funds for applied research and development in this country are relatively equal to those provided by the Government of the United States for work in that country. Each government allots approximately 0·5 per cent of the gross national product for civilian research and development. Only one other country approaches this figure, the Netherlands, where the percentage is 0·42.

At the industrial level the picture is not so bright. It is impossible to determine even approximate figures for industrial investment in research from which direct comparisons can be made between countries. There are considerable differences in the manner in which research is organized as between one country and another. For example, in this country research associations organized for co-operative research to serve a whole industry and its ancillaries play a very much greater part than they do in the U.S.A.; here there are some 40 such Associations, whereas in America there are only four or five. Generally, the American practice is either to carry out the research within an industrial organization or to have it carried out by one of the several institutions for sponsored research. Incidentally, since the war there have been developments in sponsored research in this country, and the Fulmer Research Institute and the Sondes Place Research Institute have done, and are doing, excellent work.

Another factor which makes it difficult even to make an approximate estimate of the industrial investment in research is the loose way in which the term "research" is often used. A department within an individual organization may be dignified by the title "Research" when, in fact, it is mainly, if not wholly, employed in the solution of day-to-day problems and is a "trouble-shooting" rather than an applied-research department. Such departments have great value, but the money spent on them should not be regarded as investment in research.

Industrial investment

Although, as we have said, it is not possible to determine the actual level of industrial investment in research for any country, there is one pointer which allows a reliable estimate to be made of the relative values in this country and the U.S.A. Here there are some 40 research associations carrying out work for a wide diversity of industries. They have a total income in the order of £3,400,000, of which £1,250,000 comes from public funds, and have a staff of about 4,000, of whom about one-quarter are of graduate status. Nine institutes carrying out sponsored research in America spend annually something in the order of £14,000,000, the whole of which comes direct from industry. They employ more than 1,500 qualified engineers and scientists. Furthermore, it must be borne in mind that there is much more research by individual organizations in America than in this country. Therefore, the inescapable conclusion is that industry in the U.S.A. invests much more in applied research, both relatively and absolutely, than does British industry.

Scientists' responsibilities

Scientists themselves cannot be absolved from all responsibility for industry's failure to give adequate recognition and support to applied research and development. In general, the men controlling industry are not specialists, and they cannot be expected spontaneously to recognize all the possibilities of science and technology.

It is therefore, important that scientists should be able to explain their plans, methods and results in language that can be understood by the non-specialist. Furthermore, the scientist should always bear in mind that the average industrialist is interested in applied research only in so far as it has economic possibilities. In pure research, economics can be disregarded, but in applied research every problem has its economic as well as its technical aspect. In plain words, research must be "sold" to industry.

Research programmes

That there is need for increased and better applied research and development will be generally accepted, but there are probably differences of opinion as to whether individual, sponsored or co-operative research is the best for this country. Although each has its place, in our opinion there is no doubt that the main efforts should be directed towards improving and widening the scope of research associations, particularly in the following fields:—

(a) Fundamental research on materials and processes.
(b) Applied research on urgent technical problems of general concern to the industry. This should be the largest class of work.
(c) Library and information services covering scientific and technical data and developments throughout the world.

An association dealing efficiently with the above three aspects of research work would meet the needs of the larger member firms, but probably would not satisfy those of smaller firms within the industry. These latter would be more concerned with day-to-day problems than with relatively long-term projects. It might be worth while undertaking a certain amount of *ad hoc* investigation since it should be the aim to enlist the support of all firms in the industry for which the research association caters.

Personnel recruitment

Adequate income is essential if an efficient research association is to be developed, but the recruitment of the right type of personnel is of equal importance. There is no great reserve of men with both the training and the habit of mind necessary for applied research, consequently there is urgent need to make the best possible use of those so qualified. This is one reason why co-operative research is to be preferred. There are economic reasons also, since it reduces duplication of staff, equipment and overheads, and for a given total outlay will allow the provision of larger and better equipped laboratories.

No matter what the organizational structure, the training and recruitment of the right type of personnel are matters of primary importance. For pure research a high level of academic attainment and a speculative mind are prime postulates; actual experience of the industry and its processes are secondary considerations, since the results obtained by pure research are generally of academic, and not immediate practical, interest to industry.

For applied research, however, experience of the industry and its processes should be allied to academic training and attainment. It would therefore appear that the scientist or technologist who intends to make his career in applied science should first have a certain modicum of practical experience. This would give him a clearer conception of what industry wants, which, broadly, is a development capable of relatively early application rather than an ideal solution which may be deferred to the Greek Kalends.

Training

The universities of this country are staffed and equipped to give the necessary basic training to an adequate number of engineers, but there is still room for greatly extended provision for post-graduate training schemes, such as that inaugurated by Birmingham University, specially designed for men who have spent some appreciable time in industry after graduation. It could also be an advantage if established workers in applied research were from time to time to return to industry for a refresher period.

So far as academic training is concerned, we in this country lack facilities for training technologists that are available in the United States of America. For example, there is no counterpart for the Massachusetts Technical Institute. The technical colleges in this country do good work, but in general they are organized for training technicians rather than technologists, and it is the latter that are required if the scope and amount of applied research is to be extended. There does appear to be a need for an increase in training facilities at a level between the universities and the average technical colleges in this country.

CONTENTS

COMMER TS3 DIESEL ENGINE

A 3¼-Litre, Opposed Piston, Two-Stroke Unit

IMMEDIATELY after the war it became apparent that, if the then current trends with regard to taxation of motor fuels continued, the diesel engine would become increasingly popular. Therefore, the Rootes Group decided to embark on the development of a diesel engine to provide an alternative power unit for the Commer Avenger, medium weight, passenger vehicles. The outcome is a power unit based on the rather unconventional opposed-piston type of unit. Its success during the two years or so in which it has been operated experimentally in various vehicles, including coaches running on normal routes, demonstrates the foresight and soundness of outlook of the manufacturers in undertaking such a bold venture. The design has been further proved by bench tests involving thousands of hours of running as well as in vehicles operating under the most arduous conditions at high altitudes in the Alps and on cross-country journeys in Africa. A total of over a quarter of a million miles has been covered already by test vehicles fitted with this engine.

When the unit was first conceived, the policy laid down was to produce an engine embodying the following fundamental characteristics:

1. Minimum bulk and weight
2. Low fuel consumption
3. Minimum basic cost
4. The lowest possible maintenance costs
5. A performance, when installed in Commer vehicles, equivalent to that given by the well-proved petrol engines that were already in existence.

The embodiment of all, or most, of these features is, of course, the aim of every manufacturer. Commer Cars Ltd. decided that the best possible results could be obtained only by a departure from the traditional layout of the four or six cylinder, in-line, four-stroke engine.

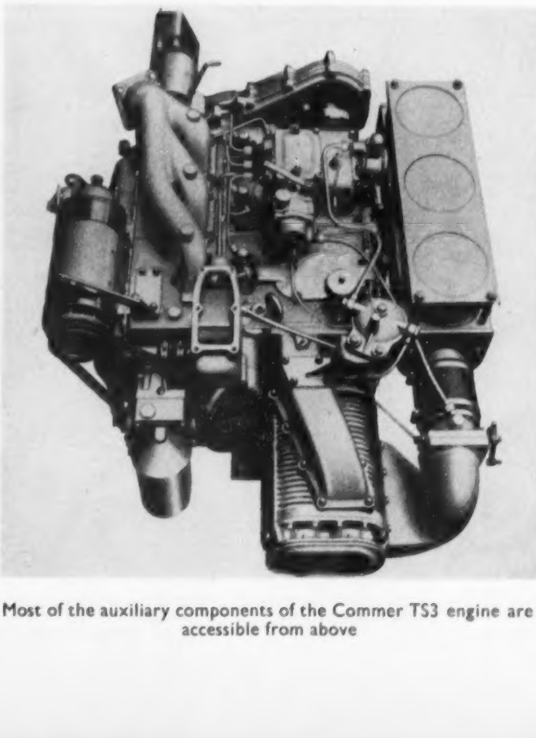
It is generally possible with two-stroke engines to obtain a better power-weight ratio than with four-stroke units, provided the two-stroke engines are of the valveless type. In addition, there is some saving so far as overall dimensions are concerned. However, in view of the fact that a blower is needed to ensure proper scavenging action over the whole operating range of speeds, it is generally thought that two-stroke units are more complex than four-stroke, even when taking into account the elimination of valves and valve actuating gear. Therefore, to produce a unit at the lowest possible basic cost, it was necessary to effect greater economies than are possible with vertical or horizontal in-line engines. It appeared that the opposed piston type of unit offered the best chance of effecting the necessary economy, since three cylinders can be employed for a power output equivalent to that of a six cylinder engine.

The advantages thus obtained are reflected in further economies throughout the whole of the design. Perhaps the most important saving is in the injection equipment. This is a conventional three cylinder type of pump serving only three injection nozzles. Single-hole nozzles are employed to reduce the likelihood of their being blocked as a result of carbon build-up. The manifolding arrangements are simplified by the three cylinder layout and so is the crankcase design. Opposed piston type engines also have advantages from the point of view of compactness. Moreover, with the layout adopted, the scope for further development is greater than would be possible with a more conventional arrangement.

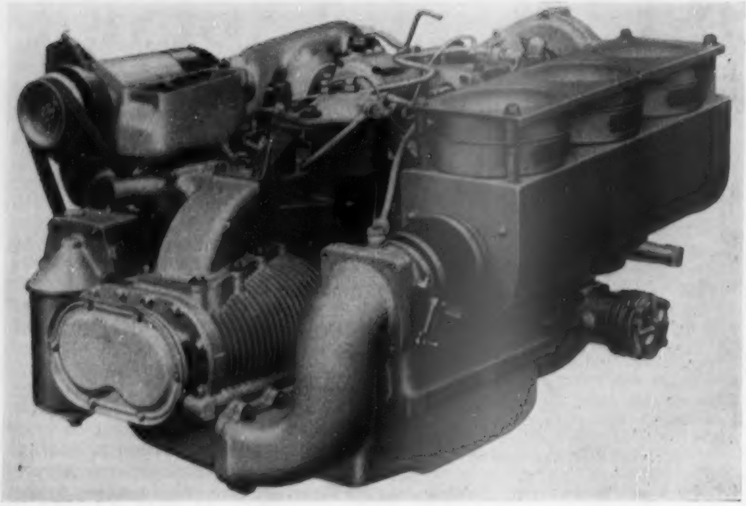
> **SPECIFICATION**
>
> *ENGINE: Three cylinders with opposed pistons. Bore and stroke 3·25 in by 4 in. Swept volume 199 in³ (3,255 cm³). Firing order: 1, 2, 3. Maximum b.h.p. 90 at 2,400 r.p.m. Maximum b.m.e.p. 94 lb/in² at 1,350 r.p.m. Maximum torque 250 lb-ft at 1,200-1,500 r.p.m. Compression ratio: 19:1 nominal, 16:1 effective. Fully balanced, four-bearing, six-throw, forged crankshaft. Kadenacy Uniflow scavenge system. Injection pump: C.A.V., N-type. Injector nozzles: single hole, biased in direction of air swirl and towards air piston. Combustion chamber type: direct injection, concave hot crowns in the opposed pistons. Fuel lift pump: Diaphragm type.*

There are also disadvantages to this layout. One is that the introduction of rockers and additional connecting links between the pistons and crankshaft adds to the complexity of the unit and partly offsets advantages gained by simplification in other respects. Moreover, the rockers are of necessity heavy, and introduce balancing problems. Balancing would, of course, present fewer difficulties were it not necessary to restrict the overall width of the units. In this engine, weights are forged integrally with the rockers to bring the centres of gravity of the rockers as close as possible to the axes of their bearings. Perhaps one of the most severe deterrents to those who have hitherto considered adopting this type of design is the difficulty of designing bearings capable of taking the heavy loads that are inevitable at the rocker pivots. The problem is complicated not only by the fact that these bearings are of the oscillating type, but also because the loading is unidirectional in the two-stroke cycle. Therefore the maintenance of a lubrication film between the load carrying faces is far from easy. That the problem has been solved successfully in this engine has been demonstrated by the satisfactory results obtained during the development period, when the engines were running under both normal and severe service conditions.

Most of the disadvantages can, of course, be overcome by employing alternative layouts, but the arrangement adopted gave the best compromise. The rocker bearings can be eliminated by employing two crankshafts per bank of cylinders, as in the Junkers aero engine, Napier-Deltic or Fairbanks-Morse units. An alternative arrangement is to employ a crankshaft with three throws per cylinder, the centre throw serving one



Most of the auxiliary components of the Commer TS3 engine are accessible from above



A large air cleaner and silencer is fitted on the left-hand, or air, side of the engine

piston and the two outer ones, with long connecting rods connected to a cross link, actuating the other piston. Examples of this type of engine are the C.L.M. Junkers slow speed types, and the Doxford marine engine.

A further unavoidable complication of two-stroke units, as compared with four-stroke types of engine, is the addition of the blower. This, unfortunately, is necessary to ensure complete scavenging. However, by using it not as a supercharger but simply to perform the charging and scavenging operations, some economy in size and power consumption has been effected. Despite these additional complications, the manufacturers state that the engine is less expensive to produce than would be a four-stroke, in-line diesel unit of equivalent performance and fuel consumption.

It seems likely, however, that this saving in cost is too small, by itself, to justify the expense involved in developing an engine that is such a radical departure from traditional conception. A more important feature is the saving in maintenance costs. These economies arise because of the relative simplicity of the injection system, the absence of valve gear requiring frequent attention during service, and the simplicity of the decarbonizing operation. The exhaust ports are so designed that deposit formation is very limited, and they are to a large extent self-cleaning. If decarbonization becomes necessary, the whole operation can be performed after the removal of only the exhaust manifold.

Among the other features that reduce servicing costs, the following two are important. Bore wear is reduced to a minimum because of the absence of heavy, piston side-thrust, which is experienced on more conventional engines in which the angularity of the connecting rod is much greater than in this unit. The journal bearings run under almost ideal conditions, because the loads on the crankshaft are almost completely in balance.

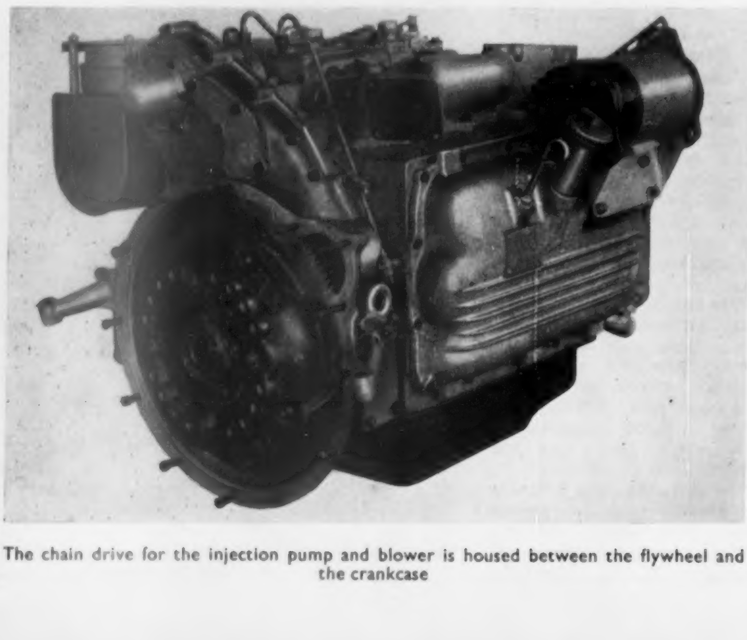
The engine is also attractive to operators because of its silence. No doubt this arises from the fact that combustion takes place more or less in the centre of the engine, instead of in a combustion chamber in which the noise is muffled only by the cylinder head casting and water jacket. The noise is different from that normally experienced in diesel engine coaches because of the sudden opening of the ports. However, the best test of noise level is the ease with which conversation can be carried on in the vehicle. Experience in coaches powered by these engines has shown that in this respect they are very good.

It is claimed that the cold weather starting of this unit is particularly easy. The manufacturers are of the opinion that this is because of its high thermal efficiency. Only a small area of the cylinder walls is in contact with the gases during combustion, so little heat is lost. The insulated crowns of the pistons further assist in restricting the heat losses.

The high thermal efficiency, together with a good mechanical efficiency, has done much towards the attainment of the low brake specific fuel consumption of the engine. The figure obtained is 0·38 lb/b.h.p.-hr and is better than that of most other two-stroke diesel units. The torque characteristics of an engine are equally important, since they determine the amount of gear-changing necessary which, in turn, has a marked influence on fuel consumption obtained on the road. As can be seen from the performance curves, the torque characteristics of this unit are good.

This is further borne out by the fuel consumption figures obtained in operation. Independent tests, of a coach of 8 tons 2 cwt gross weight, over a 35-mile undulating stretch of the London-Eastbourne road, have shown that at an average of 28·5 m.p.h. the consumption rate was 20·8 m.p.g. At a constant speed of 30 m.p.h., the consumption was 24·9 m.p.g., and at 40 m.p.h., 18·7 m.p.g.

The Kadenacy porting system has been adopted because of the good scavenging characteristics obtained with it. Its main feature is that the inertia of the outgoing exhaust gases materially assists the final stages of the scavenging operation. This characteristic is accentuated by the large cylinder length : diameter ratio of this type of engine. All the inlet ports are uncovered towards the outer end of the stroke of the pistons on the left-hand side of the engine, and are arranged so that the gas flows tangentially from a chest surrounding the ports into the cylinders. This imparts a swirling motion to the incoming gases, which are delivered by the blower and which progressively fill the cylinder and,



The chain drive for the injection pump and blower is housed between the flywheel and the crankcase

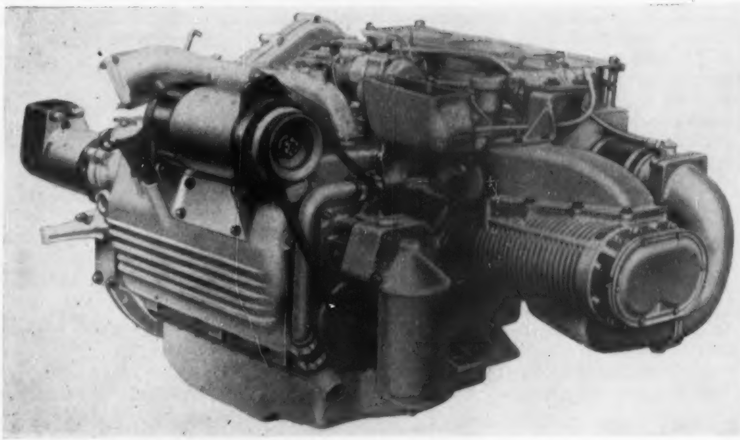
assisted by the Kadenacy effect, drive the exhaust gases out through ports uncovered by the other piston near its outer dead centre position.

There are wide differences of opinion as to the relative effectiveness of the three main scavenging systems used on two-stroke engines. These systems are, of course, the cross scavenge, the through scavenge and the loop scavenge ones. The arguments usually advanced in favour of the through, or uni-flow, scavenge system used in this engine appear to be well-founded, and most authorities agree that it is superior when, as in this engine, the cylinder length : diameter ratio is high. In theory, the principle of filling the cylinder from one end and at the same time exhausting it from the other is sound. Moreover, there is no complete reversal of gas flow as there is, for instance, in the loop scavenge system. Some authorities hold that with the Kadenacy arrangement, the swirling inlet gases are flung out by centrifugal force and leave an unscavenged core along the axis of the cylinder, but the manufacturers of this engine state that they have carried out comprehensive tests and have proved to their complete satisfaction that coring does not take place in their unit. This may be because the cylinder bore size is within a certain critical range.

A bore : stroke ratio of 0·812 : 1 has been adopted and the mean piston speed at maximum b.h.p. is 1,600 ft/min. At 1,350 r.p.m. the maximum b.m.e.p. of 94 lb/in^2 is developed. The b.h.p. developed per square inch piston area is 1·81 and the b.h.p/litre is 27·7. An engine dry weight of 997 lb is quoted, and this gives a figure of 0·09 b.h.p/lb.

The overall dimensions of the unit are: height 26¾ in, width 36½ in over the air and oil filters, and length 28¼ in. Between the front of the fan spindle and the rear face of the flywheel housing, the overall length is 44¼ in. When considering these overall dimensions, it must be remembered that the engine was designed to suit the space available in a chassis already in existence, and that different proportions could easily have been obtained. A single sandwich type rubber mounting is employed at the front and two rubber cone type mountings are fitted at the rear.

With the horizontally opposed piston type of engine, the layout inevitably differs in almost every respect from what is generally regarded as conventional. In this engine, the cylinders are horizontal with their axes positioned transversely above the crankshaft. The connecting rods are also more or less horizontal and their small ends actuate the lower ends of the centrally pivoted rockers. The upper ends of the rockers are connected to the piston rods. Bolted-on covers on each side of the crankcase give access to the whole of the mechanism so that servicing can be effected *in situ*. A sump of more or less conventional form is bolted to the lower face of the crankcase. To supply the vacuum operated brake servo system, a single cylinder, exhauster pump is mounted, with its axis horizontal in a transverse plane, on the rear of the left-hand, or air, side crankcase cover. Also on the left-hand side is the air cleaner assembly. The oil filler tube, dipstick and dynamo are on the right-hand cover.

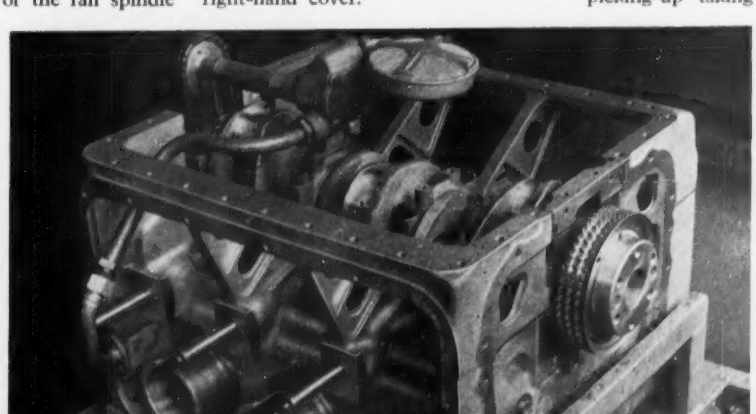


A twin V-belt drive serves the water pump and dynamo

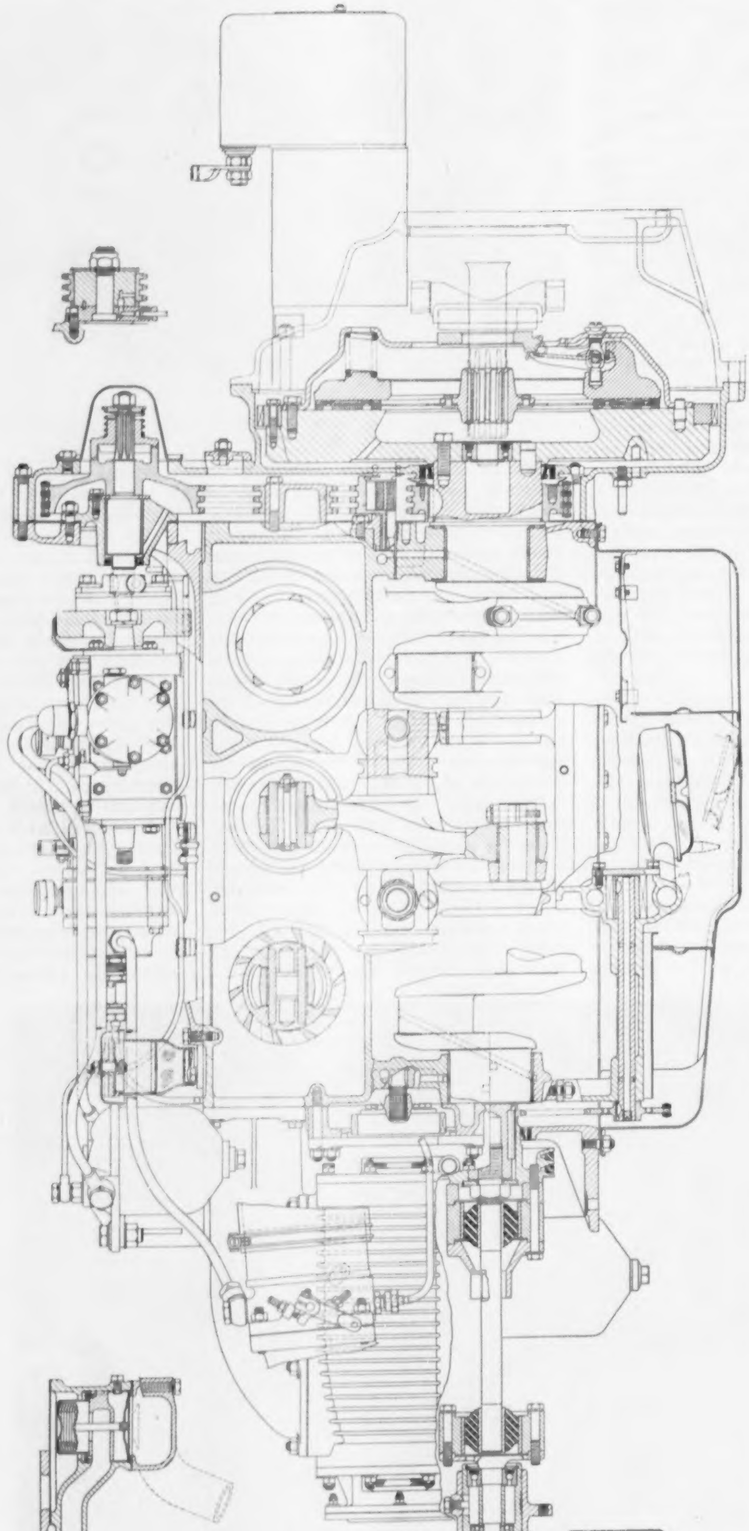
At the front of the engine are the triangulated, twin V-belt drive for the dynamo and water pump, the water outlet and thermostat, oil filter, fan drive and blower. The starter motor and the chain drive for the injection pump and blower are at the rear of the engine. A flexible coupling is splined to the rear end of a long shaft extending longitudinally through the crankcase to the blower unit at the front. On top of the engine is the exhaust manifold, the injection pump and nozzles, and fuel lift pump. Access to all these components and to most of those mounted on the sides and ends of the unit can be gained from above.

A most interesting detail feature of the engine is the exhauster pump. This is a two-stroke, single cylinder unit with an aluminium piston running in an aluminium cylinder. At first sight it may appear surprising that these two materials function together without picking-up taking place between the bearing surfaces. However, not only is the unit well lubricated by crankcase oil mist, but also there is little side thrust on the piston since its connecting rod is actuated by the lower end of the rear rocker on the left-hand side. Moreover, the high conductivity of the aluminium ensures that the heat is rapidly transferred away from the bearing surfaces.

The big end of the connecting rod of the exhauster is simply passed over the projecting end of the small end pin in the main



The crankcase casting is exceptionally light because all the loads, except those at the rocker pivot bearings, are self-balanced

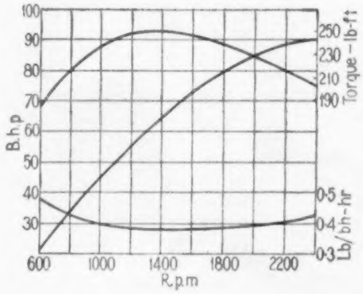


ARRANGEMENT OF THE COMMER TS3, OPPOSED-PISTON, TWO-STROKE DIESEL ENGINE
Bore and stroke 3·25 in × 4 in. Swept volume 199 in^3.

connecting rod and rocker. A $\frac{5}{8}$ in outside diameter by $\frac{3}{8}$ in inside diameter pin is employed in the compressor piston which, of course, has a stroke equal to twice the throw of the engine crankshaft. The cylinder bore is $2\frac{3}{8}$ in diameter and the unit is spigoted into the crankcase side cover and secured by four $\frac{5}{16}$ in diameter studs.

A transfer port is machined in the bore of the outer end of the cylinder, which is closed by a cast aluminium end plate, approximately $\frac{1}{4}$ in thick. This plate is secured by four $\frac{5}{16}$ in diameter studs. Ports are drilled in the piston to communicate with the transfer port at the top of each stroke; the suction port is near the bottom dead centre position in the cylinder. Thus, the action of the pump is as follows. The piston, moving from the bottom dead centre position, closes the suction port and compresses the air in the cylinder. At top dead centre, the ports in the piston communicate with the transfer port and air passes from above the crown away through the skirt of the piston. As the piston moves away from the top dead centre position, the transfer ports are closed so that the pressure in the cylinder decreases as the piston moves to the bottom dead centre position again, and uncovers the suction port, from which a pipe connection is taken to the vacuum tank of the brake servo system.

The design of a crankcase for this type of engine presents fewer problems than for units of more conventional



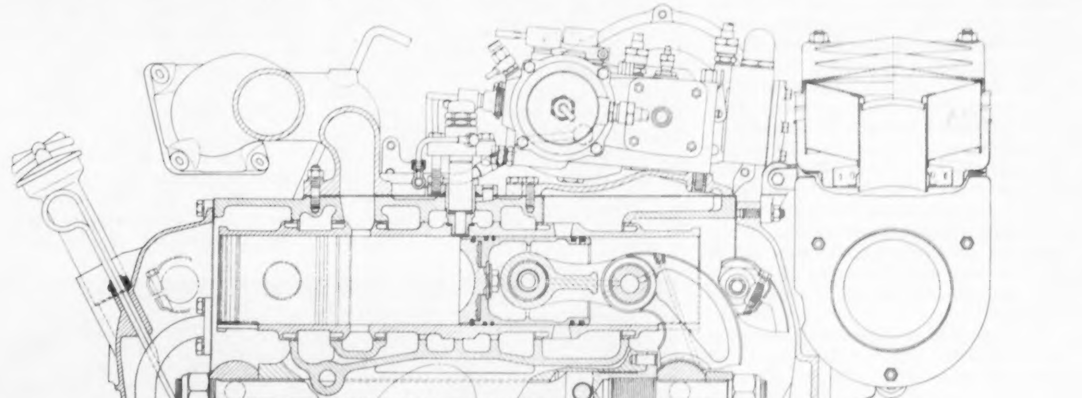
Engine performance curves

layout. This is principally because most of the loads are self-balanced, so the provision of adequate stiffness is in no way difficult. As has already been stated, the crankshaft main journal bearing loads are light. The only heavy loads that have to be taken by the crankcase are the reactions at the rocker pivots. Of these, only the loads from the pivots of the end rockers are taken through the crankcase, and these present no difficulties since they are carried by the front and rear walls. The two intermediate bearings on each side are supported by large diameter tie-bolts, which will be described in some detail later.

A simple, rectangular casting, of an iron of 14 ton/in^2 ultimate tensile strength and containing a minimum of 0·2 per cent Cr, is employed. It is supplied by Dartmouth Auto Castings Ltd. Beneath the crankcase, a face is machined to carry the sump. There is another machined face on each side for the rocker covers, and two more take the front and rear covers. The rear cover incorporates the flywheel housing. There are also machined faces on top of the unit to carry the exhaust manifold and the cover over the induction chest, on which is mounted the injection pump. All these faces are in planes at right angles to one another so that tooling for the machining operations is extremely simple. Sealing at the front cover joint is effected by a 0·008-0·012 in thick Flexoid washer. At the rear and on each side, 0·15 in thick Flexoid washers are employed, and a $\frac{1}{64}$ in thick Oakencork washer is fitted at the sump face joint.

The two intermediate webs that support the two rocker bearings on each side and two of the crankshaft main journal bearings are of light construction and are more in the nature of ties than webs. They have large holes cored in them not only to reduce weight but also to relieve casting stresses. Immediately above the rocker shafts is the box-like structure in which are carried the cylinder liners. Since wet liners are employed, coring does not present any difficulties.

The liners are supported in bores in eight webs cast longitudinally in the housing. An induction chest is formed between the two webs nearest the left-hand side of the engine. The space between this chest and another web



CROSS SECTION OF THE COMMER TS3 ENGINE

The phasing of the port opening and closing times is obtained by positioning the axis of the crankshaft slightly above the level of the centres of the lower rocker pins

An unusual feature of the two-stroke compressor unit is that its aluminium alloy piston runs in a cylinder of the same material

round the centre of each liner is divided in two by another web. The two chambers thus formed are part of the cooling jacket. There are two similar chambers on the right-hand side of the central web but the outer one is much larger than that on the inlet side and surrounds the exhaust chests. The exhaust chests for the three cylinders are not completely separated by the water jacket but are siamesed.

Brivadium liners are fitted. The sealing in the bores in the webs near each end and on each side of the inlet and exhaust chests is effected by lipped collars cast integrally round the liners and smeared with engine jointing compound during assembly. These lipped collars are an interference fit in the bores, and are of U-section to provide the necessary resilience to effect a good seal. A finish of 80 micro-in is specified for the bores of the webs and sealing faces on the lips. After assembly, each crankcase and cylinder liner unit is tested under pressure for leaks. Axial location of the liners is effected by the injection nozzle housing sleeves, which are screwed into bosses in the liners and located in holes in the top wall of the crankcase.

Because of the space requirements of the induction and exhaust chests, the distance between adjacent cylinder walls is about 2 in. The wall thickness is greatest around the combustion chamber, where it is $\frac{11}{32}$ in. Over the remainder of the length swept by the pistons it is $\frac{1}{4}$ in thick, and at the extreme ends it is reduced to $\frac{3}{16}$ in. A honed finish of 25 micro-in is specified for the bores.

The rear main journal bearing cap is extended down to the level of the lower face of the crankcase, which is 4 in below the axis of the crankshaft, and its rear face is machined to form part of the joint face for the rear cover and flywheel housing. Below the front intermediate and front bearing caps, faces are machined to receive the bolted-on casting that forms the housing for the oil pump and its drive. The seal at the front end of the crankcase is effected by the front cover, which is bolted directly to the sump joint face. Each of the four bearing caps is secured by two $\frac{1}{2}$ in diameter En 18T set bolts locked by tab washers. Lateral location of all except the rear bearing cap is effected by shoulders on each side of their seatings. The rear cap is located by a 3 in diameter dowel at the seating face, and each side of its lower end is machined to register between the flanks of the cut-out in the end wall into which it fits.

Crankshaft and flywheel

A forged En 19C crankshaft is employed. Neither the four main journals nor the three crank pins are hardened. Because the thrusts in the connecting rods are balanced and therefore the loads on the main journals are light, steel backed babbit shells are fitted. These should have a long life because of the conformability and other good bearing characteristics of babbit.

The shells are located in the usual manner by pressed-out tabs, which register in slots offset relative to one another in the abutting faces at one side of each cap and housing. All the journals are 3 in diameter: the front and rear ones are 1$\frac{3}{4}$ in, the intermediate one is 1$\frac{1}{8}$ in long. A diametral clearance of 0·002-0·004 in is specified.

From the front of the front web to the back of the rear web, the overall length of the crankshaft is 17$\frac{1}{4}$ in. The webs adjacent to each of the bearings are 0·6 in thick while those between the crank throws are 0·8 in thick. As measured across the crank pin axes, the width of the webs is 4$\frac{1}{2}$ in. The crank pins are 2$\frac{3}{4}$ in diameter.

Axial location of the shaft is effected at the rear bearing, where Vandervell semi-circular thrust washers of steel-backed babbit are carried in the ends of the bearing cap and housing. Immediately behind this journal bearing is a collar machined on the shaft. The En 9 sprocket for the chain drive to the injection pump and blower is pressed and keyed on to the 4 in diameter tail end of the crankshaft and is located against the collar. A dished thrower ring is secured by set screws to the rear face of the sprocket. It enshrouds a Gaco oil seal in a housing formed in the rear cover. The forward end of the outer periphery of the housing is lipped so that oil running down the wall of the cover is directed round it and does not run over on to the seal.

The B.S.1452, grade 17, flywheel is spigoted on to the tail end of the shaft and secured by six $\frac{7}{16}$ in diameter set bolts screwed into the shaft. Positive location is effected by two $\frac{5}{16}$ in diameter dowels. The ball bearing that carries the front end of the clutch shaft is housed in the centre of the flywheel and located between the counterbored end of the crankshaft and the circular locking plate for the bolts securing the flywheel to the shaft.

An En 18D starter ring gear with 135 teeth is pressed on to the shouldered rim of the flywheel and secured by eight $\frac{1}{4}$ in diameter set bolts. The overall diameter of the flywheel is

The rocker assemblies are readily accessible when the crankcase side covers are removed

17·08 in with the ring gear and 16·45 in without. It is 1·90 in thick, weighs 102 lb and its mass moment of inertia is 4,130 lb/in^2.

Connecting rods and rockers

The connecting rods are of forged En 16. Their centre-to-centre length is 8·11 in. They are of H-section, the minimum dimensions of which are: $1\frac{1}{4}$ in deep, 0·85 in wide over the flanges by $\frac{7}{32}$ in web thickness. Each rod weighs 4 lb. Since there is no question of withdrawing the rods througn the cylinder bores as in more conventional layouts, their big ends are split at 90 deg. They are located by the fit of the $\frac{3}{8}$ in diameter, En 18T, waisted holding-down bolts, which are locked by tab washers. The waisting is effected, of course, to increase the resilience of the bolts and to avoid stress concentration at the threaded end.

Vandervell thin-wall bearing shells are fitted in the big ends. The rod halves are lined with lead indium and the caps with babbit. This arrangement has been adopted to take advantage of the conformability and embeddability of the babbit material, while at the same time using the harder bearing material in the highly loaded areas. The use of a different lining material for each half of the big end bearing shells is particularly suitable for two-stroke engines because of the relatively constant loading on the rod half. However, it is of interest to note that the arrangement is also employed on at least one high performance four-stroke petrol engine currently in production in this country.

The nominal inside diameter of the big end bearings is $2\frac{3}{4}$ in, and their length is $1\frac{1}{8}$ in. A diametral clearance of 0·002-0·0035 in is specified. Location is effected in the same way as that described for the main journal bearing shells.

Unlike conventional connecting rods, these have forked small ends which carry the pins that connect them to the rockers. Each blade of the fork is about $\frac{11}{16}$ in thick and its eye is $2\frac{1}{8}$ in mean outside diameter by $1\frac{3}{8}$ in inside diameter. The eyes are split and fitted with $\frac{1}{4}$ in diameter set bolts to clamp the small end pins. These pins are $1\frac{3}{8}$ in outside diameter by $\frac{11}{16}$ in inside diameter and they are of Nycrocase, or En 36 steel, carburized and hardened to a minimum of 58 Rockwell C.

Many problems had to be faced in the design of the rocker assemblies and there are few people with experience of the application of this type of mechanism to diesel engines. However, it is surprising how little development work is needed to perfect new mechanisms, provided well proved fundamental principles are applied initially. Evidence of the care with which every detail has been considered is given in the illustration of the cross section of the engine, which shows the arrangement for tying the two shafts together with bolts passed transversely through the crankcase.

The assembly on each side comprises the $2\frac{1}{4}$ in outside diameter by $\frac{3}{4}$ in inside diameter En 361 shaft, the three En 16T rockers and two cast iron spacer rings. A spacer ring is fitted in front of each of the two rear rockers but not with the front rocker. The two rocker and ring assemblies and the front rocker are each located between the machined end faces of pads on which is seated the rocker shaft. These pads are cast integrally with the crankcase.

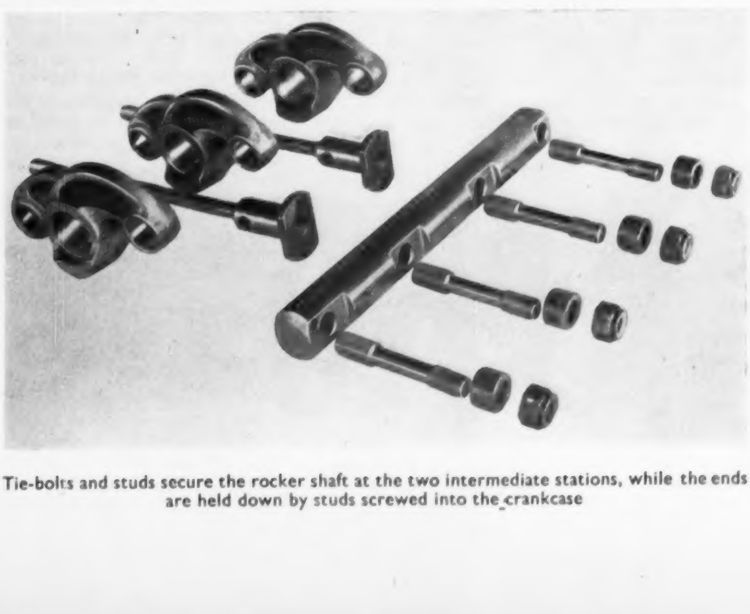
There are four holding-down points on each shaft, and six studs and two tie-bolts are employed. Both of the tie-bolts and two of the studs are used at the intermediate stations, that is, in front of and behind the two centre rockers. The four remaining studs secure the ends of the shafts. The tie-bolts are threaded $\frac{7}{8}$ in diameter at their right-hand ends. They are waisted to $\frac{3}{4}$ in diameter for a length of just over 12 in. At the other end of each, there is a 1 in fillet radius between the waisted portion and a $1\frac{1}{2}$ in diameter head which is about 3 in long. The head is of cylindrical section and its periphery is undercut for a length of about $1\frac{1}{2}$ in approximately midway between its ends. This undercut reduces the amount of machining to be done to close tolerances on the periphery. A flange is formed round the outer end of the head which is counterbored 1 in diameter and tapped to receive two of the six holding down studs.

Each tie-bolt is passed from the left through a clearance hole in the crankcase, which is counterbored to receive the head. It is held in position by a set-screw passed through a countersunk hole in the flange into a tapped hole in the crankcase and is positively located by a $\frac{3}{8}$ in diameter dowel in a hole diametrically opposite the set-screw. Positive location is necessary because the blower drive spindle passes through holes drilled diametrically through the heads of the tie-bolts. Details of this drive arrangement are given later.

Four clearance holes are drilled diametrically through each rocker shaft to receive the ends of the studs and tie-bolts, and flats are machined on the shaft round the ends of the holes. The flats on one side of the shaft seat on the pads that locate the rockers; self locking nuts, together with distance pieces on the end studs, are pulled against those on the other side. When the tie-bolts and studs have been assembled into the crankcase, the right-hand rocker shaft is mounted on them and tightened down on to the pads. The tension in the tie-bolts and studs is controlled by tightening the nuts with an adjustable torque wrench set to slip at 160 lb/ft.

The studs that are fitted in the tapped counterbores in the ends of the heads of the tie-bolts are identical with those that are screwed directly into the crankcase to secure the ends of the shafts. All are of En IIIT and are waisted to $\frac{3}{4}$ in diameter. At their outer ends they are threaded $\frac{7}{8}$ in diameter. The left-hand rocker shaft is assembled over the studs and secured in the same manner as that on the right, except that distance pieces, $1\frac{1}{2}$ in outside diameter by $\frac{7}{8}$ in inside diameter by 1 in long, are interposed between all the nuts and the shaft, instead of on the end pair only. These distance pieces are of En 32B and are cyanide hardened to prevent the nuts from bedding in and some of the tension in the stud thereby being lost. The distance pieces are necessary because, if they were not fitted, the studs would be relatively short and would have insufficient resilience. Lack of resilience would result in appreciable loss of tension in the bolt if even the slightest degree of yield were to take place in service.

The centres of the bearings in the ends of the arms of each En 16T rocker are offset $1\frac{1}{8}$ in relative to one another. This is because the centres of the crank throws carrying the opposed connecting rods that serve common cylinders are spaced $2\frac{7}{32}$ in apart. Clevite 10 bearing bushes of generous proportions are pressed into the centre of each rocker and a balance mass is incorporated on the boss. They are $2\frac{11}{16}$ in long by $2\frac{1}{4}$ in inside diameter. The running clearance is 0·002-0·0038 in and the



Tie-bolts and studs secure the rocker shaft at the two intermediate stations, while the ends are held down by studs screwed into the crankcase

interference fit of the bushes in their housings is 0·004-0·007 in. Balance weights are forged integrally with the bosses that form the bearing housings to effect the greatest possible degree of balance.

Inside the bush, that portion of the bearing surface which takes the thrust loading has two longitudinally machined, $\frac{1}{4}$ in wide spreader grooves in it, spaced about 54 deg apart. Further distribution of the oil is effected by a groove of much smaller section. This groove is shaped something like a figure of eight, the boundary lines of which are continuous, but which do not cross at the centre. It is superimposed on the other two and the major axis of the figure is at right angles to those of the other two grooves. The general principle that has been followed in the design of this oscillating bearing bush is that the grooves should be spaced in such a manner that together they wipe the whole of the area of the shaft where thrust is taken. In this way the oil is spread from the grooves over the shaft and a continuous film is maintained between the bearing surfaces, despite the fact that a good hydrodynamic wedge effect, such as is obtained with rotating shafts, cannot be obtained with oscillating shafts.

At each end of the rocker, a Clevite 10 bush is pressed in. The bushes are 1$\frac{3}{8}$ in long by 1$\frac{1}{8}$ in inside diameter. Their interference fit in the housings is 0·008-0·005 in and the running clearance is 0·0023-0·001 in. The En 36 pins are 1$\frac{3}{8}$ in outside diameter by $\frac{11}{16}$ in inside diameter, but whereas the length of that at the connecting rod end is 2$\frac{7}{8}$ in, the one at the piston rod end is only 2$\frac{7}{16}$ in long. The difference in length is accounted for by the fact that the cross section of the eyes in the forked end of the piston rod is appreciably smaller than that in the connecting rod.

That this should be so is at first sight surprising, since a rocker ratio of 1 : 1 has been adopted. However, a careful analysis of the loads in the mechanism shows that the eye with the largest cross section is, in fact, the most heavily loaded one, and that its loading is mainly tensile and therefore more critical so far as fatigue is concerned. The pins at the lower ends of each rocker are positively located by the pinch bolts registering in grooves in their peripheries. At the piston rod end, the pin is sealed, to prevent oil leakage, by conical mild steel plugs secured by a tie bolt passed through its bore.

Piston rods and pistons

The piston rods are made of En 16T. Their centre-to-centre length is only 3$\frac{3}{4}$ in, therefore the ends of the cylinder liners have had to be slotted to clear the rocker arms as the pistons move to inner dead centre. The rods are of H-section, the minimum dimensions of which are 1$\frac{5}{16}$ in over the flanges by $\frac{1}{2}$ in web thickness.

At the outer end, each rod is forked in a similar manner to the connecting rod and the eye in each fork is split and clamped round the pin by a pinch bolt. A single eye at the other end of the rod carries the pressed-in, phosphor bronze bush for the gudgeon pin. This bush is 1$\frac{3}{4}$ in long by 1$\frac{3}{8}$ in diameter.

An LK5 gudgeon pin is employed; it is nitrided to a minimum Vickers hardness value of 900. Axial location of the pin is effected by brass end pads. The counterbored shanks of the end pads are about $\frac{7}{8}$ in long and are a 0-0·001 in interference fit in the bores of the gudgeon pin. The finish of the bores over a length of about 1 in from each end is 80 micro-in, to avoid damage to the relatively soft material of the end pads if they move in their bores while the engine is running. The effective bearing length of each end of the gudgeon pin in the piston boss is $\frac{5}{8}$ in.

Among the difficulties that have to be faced when designing pistons for two-stroke engines is the solution of problems concerning heat flow. These arise from the fact that there is a firing stroke every revolution. In many designs, oil cooling is employed to carry away the heat. However, in this engine the principle adopted has been to insulate the crown from the remainder of the piston and so retain the heat in the end adjacent to the combustion chamber. This is a sound arrangement, since it helps to improve thermal efficiency and to reduce diesel knock.

The pistons, supplied by Hepworth & Grandage Ltd. are of composite construction. A cast iron body is employed and the crown is of KE 965 steel, commonly used for poppet valves. The periphery of the crown is spigoted into the end of the skirt component, and a boss at its centre projects through a hole in the skirt. This boss is 1 in diameter for a length of about $\frac{1}{4}$ in and then is reduced to $\frac{7}{16}$ in diameter and threaded for a nut which, together with a steel washer of the Belleville type, secures the crown to the skirt.

Interposed between the crown and the skirt is a cast iron bearing plate. On the base of the crown, where it is pulled against the bearing plate, are two circular ribs, one round its outer periphery and the other between it and a shoulder round the boss in the centre. These ribs and shoulder form the only contact between the crown and the bearing plate so that the path through which heat can flow through the skirt is restricted. The undersurface of the bearing plate is pulled against a flat seating machined on the end of the skirt component. The periphery of the crown is shouldered above the spigot and a fire ring is accommodated in the groove thus formed between the crown and the skirt.

In section, the fire ring is of crank-handle form. The crank fits into the groove and carries a rectangular sealing ring. The remainder of the section fits round the crown. Beneath this are two compression rings of wedge section. When the assembly is in the cylinder, their gap is 0·009-0·014 in. The face width of both rings is 0·125-0·124 in and their radial thickness is 0·116-0·108 in. In the groove, the side clearance is 0·001-0·0015 in, and the depth of the groove is 0·132 in.

Two more rings are fitted round the base of the skirt. The top one is a sealing ring and is of L-section, fitted with the foot of the L uppermost. The second is a two-piece slotted K.S.S. scraper ring, of the type described in the February 1954 issue of *Automobile Engineer*. These rings are both pegged and their working gap, in which the $\frac{3}{16}$ in square section peg is accommodated, is 0·202-0·197 in. The face width of the sealing ring and of each of the two parts of the oil scraper ring is 0·039 in and the radial thickness of each component is 0·118-0·128 in. Between the rings and their grooves, the side clearance is 0·001-0·003 in and the depth of the grooves is 0·160 in.

Blower and injection pump drives

The injection pump and blower are chain driven. The drive is enclosed by

An unusual form of grooving is employed in the rocker pivot bearings

The ends of the cross holes in the piston are sealed with brass plugs

a STA7AC4, or LM4M, dished cover, on the rear of which is incorporated the flywheel housing. This cover is bolted to the rear face of the crankcase and to the continuation of this face on a bracket on top of the crankcase. The bracket carries the bush type bearings for the injection pump drive spindle.

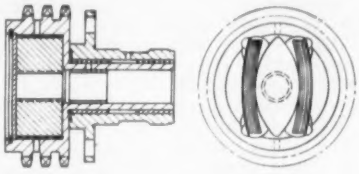
An En 9 sprocket, on the rear end of the crankshaft, drives the $\frac{3}{8}$ in pitch three-strand chain. The axis of the injection pump sprocket is about $15\frac{1}{2}$ in above and to the left of that of the driving sprocket. Closer to the driving sprocket, with its axis about $5\frac{3}{4}$ in away is the blower drive sprocket. Vertically above the blower drive sprocket is a jockey sprocket meshing with the outer face of the chain. This jockey sprocket serves to wrap the chain round the blower drive sprocket. Another jockey sprocket meshes with the inner face of the chain on the other side. It is approximately midway between the meshing points of the driving and injection pump sprockets and is eccentrically pivoted so that adjustment to the chain tension can be effected. Access for this adjustment is gained through a cast aluminium plate bolted on to the rear cover.

The teeth of the blower drive sprocket are machined round an En 9S cup-shaped component. A 1 in outside diameter by $\frac{23}{32}$ in long tubular extension of the base of the cup is carried in two $1\frac{3}{16}$ in long phosphor bronze bushes, spaced $\frac{13}{16}$ in apart. These bushes are pressed into a flanged, cast iron housing, which is spigoted into a hole in the rear wall of the crankcase. The assembly is secured by three bolts passed through holes in the flange.

A spring drive arrangement is housed in the sprocket cup and retained by a Seeger circlip and steel disc just inside its rim. The assembly inside the cup comprises two sets of leaf springs between which is an elliptical head splined on the end of the blower drive spindle. When the drive is not being taken, the minor axis of the ellipse is perpendicular to the leaf springs but, when the drive is taken up, the major axis of the elliptical head tends to move round to the position previously occupied by the minor axis. The springs resist this action and so the drive is cushioned.

Immediately in front of the spring drive, the En 9T spindle is carried in a $\frac{5}{8}$ in diameter bush in the tubular extension of the cupped sprocket. Forward of the bush, the spindle is waisted to 0·485 in diameter and extends the whole length of the crankcase and through the clearance holes drilled diametrically in the heads of the two transverse tie bolts that help to secure the rocker shafts. The waisted portion of the spindle is approximately 20 in long so it helps to make the drive even more flexible. At the front, the end of the spindle is splined into a gear on the rear of the blower unit. This gear meshes with a pinion, and drives the blower at 1·8 times engine speed.

The injection pump drive is carried by a bracket formed by a vertical extension of the cast aluminium top cover, which closes the front of the rear cover above the level of the top face of the crankcase. The En 351 spindle is carried in two 0·9 in long by $1\frac{1}{4}$ in inside diameter Vandervell bi-metal bushes, spaced $\frac{3}{16}$ in apart in a flanged cast aluminium housing spigoted into a hole in the closing plate and secured by six $\frac{5}{16}$ in diameter studs and nuts. In the front end of the housing is a lip type oil seal that bears on the spindle, which at that point is reduced to $\frac{7}{8}$ in diameter.


The spring drive unit for the blower

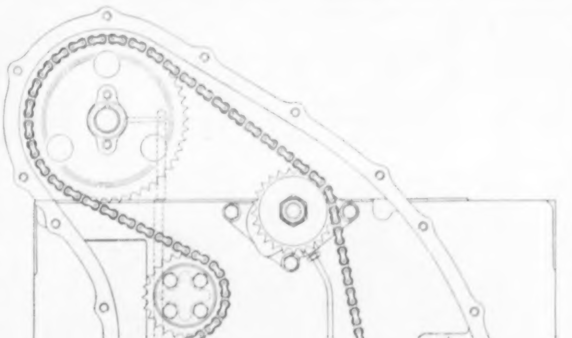
A circular thrust plate is bolted to the rear face of the housing and registers between a shoulder on the spindle and the front of the boss of the sprocket. Where the spindle passes through the thrust plate, it is $1\frac{1}{16}$ in diameter for a length of $\frac{1}{8}$ in. It is then reduced to $\frac{3}{4}$ in diameter to carry the En 9S sprocket, which is floating on the spindle. This sprocket is retained by a circlip, and it forms one member of a dog clutch by means of which it is driven. The other member of this clutch is splined on to the end of the spindle and held in engagement by a coil spring in compression between it and a retainer washer pulled against a shoulder of the spindle. The whole assembly is retained by a nut on the end of the spindle.

The end faces of the dogs are inclined, and if the engine starts running in reverse, the dogs ride up the inclines until the timing is 140 deg out of phase. This effectively prevents running in reverse for more than a couple of revolutions at the most. When the engine is turned in the normal direction of rotation again, the clutch slides back into correct engagement. Access to the clutch is gained by removing a cupped steel pressing which is bolted over an aperture in the rear cover, through which the assembly projects.

Injection equipment

A C.A.V. flexible coupling, incorporating the means of timing adjustment, is interposed between the spindle and the C.A.V., N-type pump. This pump is secured by four set bolts to a cast aluminium cover bolted over the induction chest in the crankcase. A pneumatic governor is employed and the usual excess fuel device for starting is incorporated.

The injection pump is served by a conventional diaphragm type fuel lift pump mounted on the front of the crankcase. C.A.V. injection nozzles, each with a single 0·55 mm diameter hole, are employed. The operating pressure is 140 atmospheres, and the injection timing is 28 deg before inner dead centre.


An eccentric, chain tensioner sprocket as well as a simple jockey sprocket are employed in the drive for the injection pump and blower

Blower, and induction and exhaust systems

The Roots type, three-lobe blower, supplied by Wade Engineering Company Ltd., is driven at 1·8 times engine speed. The maximum pressure it delivers is 6 lb/in^2. It is not a pressure charger, but is intended only for scavenging purposes. At 2,400 r.p.m. it delivers 430 ft^3/min and absorbs 13 b.h.p.

One or two features of the blower are of particular interest. Labyrinth seals are incorporated at the ends of the rotors to reduce friction losses. To minimize noise, the outlet port is positioned obliquely relative to the axes of the rotors. Lubrication is effected through ducts drilled in the casing; this arrangement is described in more detail under the heading "Lubrication."

Air is drawn through three oil bath air cleaners mounted on a single, tuned

silencer. The whole cleaner and silencer unit is supplied by Burgess Products Ltd. A rubber hose connection is interposed between the silencer and the casting carrying the butterfly valve. This casting is flanged and bolted to the blower intake, and a connection is taken from it to the pneumatic governor.

From the blower outlet, the air is passed to the induction chest. Thence it goes through the tangential induction ports to the silencers. The inlet ports open 43 deg before outer dead centre and close 56 deg after outer dead centre; the exhaust lead is 25 deg and the air lap is 1½ deg. Therefore, the exhaust ports open 68 deg before outer dead centre and close 54½ deg after. The phasing has been obtained by positioning the crankshaft axis slightly high relative to the centres of the lower pins.

The exhaust manifold is of STA7 AC4 cast aluminium and is fitted with Cooper joint washers. This weight-saving is possible because the exhaust temperature is relatively low. Each branch pipe of the manifold is 3 in diameter and the tail pipe is tuned to assist the scavenging operation at cruising speeds, that is, at 1,400 to 1,500 r.p.m. This tuning effect should not be confused with the Kadenacy effect.

According to Schweitzer, in *Scavenging of Two-Stroke Cycle Diesel Engines*, published by MacMillan and Co. Ltd., New York, the Kadenacy effect is defined as the assistance given to scavenging by the inertia of the gas column, not in the exhaust pipe but in the cylinder. He says, however, that gas inertia in the exhaust pipe can be of powerful assistance to the Kadenacy effect if the pipe is properly tuned. In order to take full advantage of this effect, the exhaust lead must be large enough to permit the gas to discharge from the cylinder and create a depression in its wake before much air is admitted through the inlet ports, but not for so long that the return wave from the exhaust pipe would reach the exhaust ports while they are still open and re-admit burned gases to the cylinder.

A rapid exhaust opening is another essential and is the reason why in this engine the exhaust ports occupy a large proportion of the circumference of the cylinders but only a small proportion of its length. Efficiency can be further increased by streamlining the ports. In the Commer engine, the inlet ports converge slightly and the exhausts diverge. A good feature of the Kadenacy system is that as the load increases, the back pressure in the induction system, and therefore the power consumption by the blower, decreases. In industrial engines, running at constant speed, a blower would be unnecessary provided the exhaust pipe were properly tuned. However, in this commercial vehicle application, the blower is used to give efficient scavenging over the whole range of operating speeds.

Fan, water pump and cooling system

In this particular installation, the fan and radiator are mounted on the frame, independently of the engine. Therefore, the fan is driven through two flexible couplings on the ends of a ¾ in diameter by 9 in long spindle. The rear coupling assembly, which is bolted to the V-belt pulley, comprises four components. One is a distance tube which is spigoted on to the pulley. This distance tube holds the remainder of the assembly clear of the set bolt and washer that secure the pulley to the crankshaft. Spigoted into the distance piece is a housing for the flexible unit. This unit comprises two coaxial tubes between which is bonded a rubber bush; the outer tube is pressed into the housing and the inner one is pressed on the shaft. A cranking sleeve, spigoted into the front end, forms the fourth part of the assembly. It has a hexagonal periphery so that it will take a spanner by means of which the engine can be turned for servicing operations. The whole assembly is held together by three 5/16 in diameter set bolts screwed into the pulley.

The flexible coupling at the front end of the shaft is similar, except that there is no cranking sleeve or distance piece. The housing is secured by set bolts to a flange on the rear end of the fan spindle. On a ⅝ in diameter portion near the front end of the spindle, the cast iron boss of the 16½ in diameter, five-bladed, pressed steel fan is keyed on. It is retained by a nut on the ½ in diameter threaded end of the spindle.

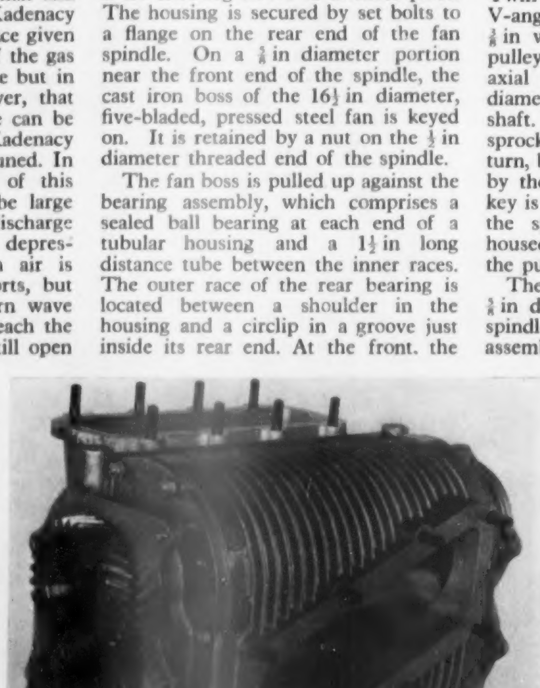
The fan boss is pulled up against the bearing assembly, which comprises a sealed ball bearing at each end of a tubular housing and a 1½ in long distance tube between the inner races. The outer race of the rear bearing is located between a shoulder in the housing and a circlip in a groove just inside its rear end. At the front, the other outer race is not positively located axially. The tubular housing is carried in a flanged boss bolted to a bracket on the radiator mounting. Location against rotation in this boss is effected by a set bolt, the dowel end of which registers in a slotted hole in the tube. The major axis of the slot is parallel with the axis of the spindle so that the whole assembly has limited freedom to float axially.

This dog clutch in the injection pump drive prevents running back. A shaft is shown separately below it

As has already been stated, the drive for the water pump, which operates at 1·47 times engine speed, is effected by a triangulated V-belt arrangement. Twin V-belts are employed. Their V-angle is 40 deg and they are each ⅜ in wide by 5/16 in thick. The drive pulley is secured by a set bolt in an axial hole in the end of the 1⅝ in diameter front extension of the crankshaft. It is pulled against the drive sprocket for the oil pump which, in turn, bears against the shoulder formed by the front main journal. A single key is furnished to drive the pulley and the sprocket. A lip type oil seal, housed in the front cover, bears round the pulley boss.

The driven pulley is pressed on the ⅝ in diameter front end of the pump spindle. This spindle is part of an assembly supplied by the Hoffmann Manufacturing Co. Ltd. It comprises two ball bearings in a tubular outer race and running in grooves round the spindle, and a seal in each end. The whole assembly is carried in the STA7AC4 aluminium alloy nose piece of the pump casing. It is secured by a dowel ended set screw in a radially drilled and tapped hole in the nose piece; the dowel end registers in a hole in the tubular outer race. The nose piece is secured to the body of the pump, which is also of STA7AC4, by five ¼ in diameter studs and which is bolted to the front end of the crankcase. Both the nose piece and the body are anodized after bright dipping.

Immediately to the rear of the bearings, the shaft is reduced to ½ in diameter. A



The blower outlet port is arranged obliquely to reduce noise

rubber thrower ring is assembled up to the shoulder thus formed and is located in a groove round the spindle. Behind the thrower is a spring-loaded water seal with a moulded-in carbon thrust ring. This seal is housed between the rotor and a washer that fits against a shoulder in the bore of the nose piece. The seal does not rotate, and the carbon thrust ring bears on the rotor. A drainage hole is drilled in the base of the nose piece between the water seal and the bearing assembly. The 3·23 in diameter rotor is pressed on the rear end of the spindle. It is of a cast iron, the ultimate tensile strength of which is 12 ton/in^2.

From the pump the coolant passes into the front end of the block, on the exhaust side. After flowing round the exhaust chest, it passes more or less axially along the outside of each cylinder liner. In the webs that support the cylinders, the slots through which the water passes are cut at 180 deg to one another to ensure complete circulation round the liners. The water outlet is adjacent to the blower inlet at the front end of the block. It houses a thermostat, which begins to lift at 169 deg F and is fully open at 196 deg F.

A gilled tube radiator with tubes of elongated cross section is employed. Its frontal area is 425 in^2 and the block thickness is $2\frac{1}{4}$ in. The system is pressurized to 4 lb/in^2.

Oil pump and lubrication system

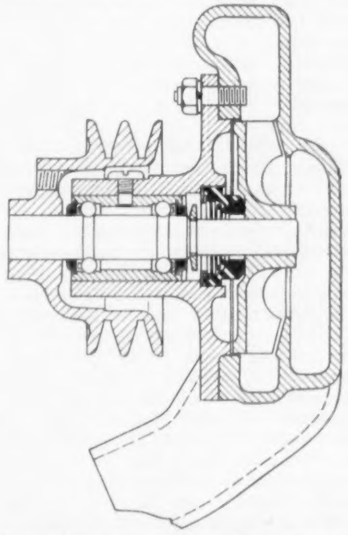
The gear type oil pump is mounted with the axis of its drive spindle parallel to that of the crankshaft. It is in a cast iron housing bolted to Nos. 1 and 2 main journal bearing caps, the lower faces of which are machined to receive it. In this position, a small part of the base of the gear housing is submerged below the sump oil level. The reason why it is not totally submerged is that were it so, the drive sprocket and chain at the front end would also be submerged. This sprocket is of En 8Q, and is pressed and keyed on to the spindle. The driving sprocket on the crankshaft is also of En 8Q, and the drive ratio is 1:0·84. A $\frac{3}{8}$ pitch chain is employed.

A cyanide hardened, En 32A driving spindle is employed. Its outside diameter is $\frac{5}{8}$ in and it is drilled axially $\frac{1}{4}$ in diameter for lubrication purposes. The front end of the axial hole is sealed with a screwed-in plug. Oil enters at its rear end, which is inside the pump chamber, and is passed along the hollow shaft and through radial holes to the two bearings bored directly in the casting. The front bearing is $1\frac{5}{8}$ in long and that at the rear 2 in long. They are spaced about 3 in apart. Between them, the top of the casting is open so that it forms a trough to receive oil splashed from the crankcase further to assist in lubricating the bearings.

Keyed on to the spindle at its rear end, which is beneath the front intermediate, crankshaft bearing, is the $1\frac{5}{8}$ in long En 7 pump gear. The idler gear is of cast iron and is carried on a $\frac{5}{8}$ in diameter En 1A spindle pressed into a boss in the housing. The gear assembly is retained by a $\frac{1}{4}$ in steel plate bolted on the rear end of the housing.

The sump capacity is 28 pints. Oil is drawn through a floating pick-up incorporating a gauze strainer of approximately 50 in^2 area, and thence into the pump inlet, which is cast in the housing and is below the oil level. From the pump outlet, a $\frac{3}{4}$ in outside diameter pipe carries the lubricant inside the crankcase to a connection on the right-hand side of the front end of the cylinder block. Drilled ducts then pass it through an A.C., full-flow filter. In the head casting of this filter is incorporated an adjustable relief valve which blows off at 55-65 lb/in^2. Provision is also made in the head casting for connections to an oil cooler, for use when the engine is employed for stationary applications.

From the relief valve, oil is directed into the $\frac{3}{8}$ in diameter main gallery drilled longitudinally in the right-hand side of the crankcase. Lateral passages, $\frac{1}{4}$ in diameter, serve the main journals. The crankshaft is drilled in the usual way between the main journals and the crank pins to serve the big ends. These holes are $\frac{7}{32}$ in diameter.



Many components of the water pump are common with those in units fitted to other Commer engines

The hollow rocker shafts are supplied with oil through two, $\frac{3}{8}$ in diameter transverse passages drilled from the main gallery. One is at the front and serves the right-hand shaft, and the other is at the rear and feeds the oil to the left-hand shaft. At the outer ends of these transverse holes, the oil enters the shafts through hollow dowels.

From these dowels, the lubricant is passed along the length of the $\frac{3}{4}$ in diameter hole in the shaft and round the waisted studs. Two radial drillings serve each rocker bearing. From the grooves in the bore of each bearing, radial holes distribute the lubricant into an annular groove in the housing, and thence through $\frac{3}{16}$ in diameter holes to the bushes at the ends of the rockers. Star-grooving in the bores of these bushes spreads the oil from these holes over the bearing surface.

An intermittent feed is supplied to the gudgeon pin bush in the following manner. As the pin at the outer end of the piston rod oscillates with the rod, radial holes in it connect the feed hole in the rocker with an axial drilling in the piston rod. This axial hole passes the lubricant along the rod to the small end bush. The bore of this bush is helically grooved, and the grooves are so spaced that during each oscillatory motion of the bearing they together wipe the whole of the periphery of the gudgeon pin.

Oil is passed to the chain drive at the rear end of the engine through a $\frac{3}{8}$ in diameter passage drilled transversely from the main gallery. A $\frac{3}{16}$ in outside diameter pipe is connected to this transverse passage and feeds oil into the meshing point between the chain and the crankshaft sprocket. Another pipe of the same size serves the chain tensioner sprocket. There is a vertical drilling $\frac{3}{16}$ in diameter from the left-hand end of the transverse one. Branches from this vertical drilling serve the jockey sprocket and the injection pump drive-sprocket bearings.

The transverse drilling breaks into the hole in the crankcase, in which is contained the housing for the bearings that carry the tubular extension of the cupped sprocket of the blower flexible drive unit. There is an annular groove round this housing and a radial hole feeds the lubricant from the groove to the space between the two bearing bushes. Inside the front bush, another hole is drilled radially through the tubular extension and through the other bush inside it, which carries the spindle that drives the blower.

At the front end of the engine, the blower is served from the transverse drilling that feeds oil into the right-hand rocker shaft. This transverse drilling has an elevated overflow at its outer end so as to maintain only a small head of oil to serve the blower. A longitudinal passage is drilled from the crankcase, through the blower casing to the front end bearing. A return passage drains the lubricant back into the gear casing at the rear of the blower unit, where the level is retained by a weir. The overflow from this weir drains to the sump.

Military Aircraft

THE complete range of shore-based and ship-borne aircraft demanded in modern war are reviewed in " Military Aircraft of the World," by H. F. King, M.B.E., published by Iliffe and Sons Ltd., Dorset House, Stamford St., London, S.E.1, at 2s. 6d. net (postage 4d.). Many of the numerous illustrations appear for the first time. There is a spread of drawings depicting the world's jet bombers.

TIN-NICKEL ALLOY PLATING

A New Decorative and Protective Coating

ELECTRO - DEPOSITED tin-nickel alloy plate is a completely new decorative and protective coating, which has been developed in the laboratories of the Tin Research Institute, Fraser Road, Greenford, Middlesex. When plated on a polished basis metal, the alloy deposits in a bright form and little or no finishing is required. The most outstanding property of the deposited plate is its resistance to tarnishing, in which respect it is as serviceable as chromium. It also has a more attractive colour than either nickel or chromium; the blue and yellow tints respectively associated with these metals are absent, the tin-nickel alloy plate having a faint rose-pink tint that is generally regarded as pleasing and attractive. The plate is harder than either tin or nickel and is considerably more resistant to attack by many corrosive media.

The alloy deposited by the process described below contains approximately 65 per cent tin and 35 per cent nickel. Unless the electrolyte or plating conditions are hopelessly out of balance, the composition of the deposit never varies by more than a few per cent. These figures correspond quite closely to an equi-atomic ratio, and experience shows that the process favours the co-deposition of tin and nickel atoms at identical rates.

Tin and nickel are both fairly resistant to corrosion, but each metal tarnishes in the air and the initial brilliance obtained by polishing or other means is soon lost. However, tin and nickel form an alloy series characterized by a number of inter-metallic compounds, which can exhibit properties entirely different from those of their constituent metals. By electro-deposition, it is possible to co-deposit tin and nickel in the form of a compound containing approximately equal numbers of tin and nickel atoms; the alloy deposit obtained possesses some unique properties.

The tin-nickel alloy may be deposited direct on copper and copper-base alloys, but when steel is the basis metal a copper undercoat is recommended. Because the tin-nickel electrolyte attacks aluminium and zinc-base alloys, these metals cannot be plated direct but must first be protected by plating with a pore-free deposit of copper or alternative metal. For most purposes a deposit thickness of 0·0005 in (0·013 mm) will meet requirements. For purely decorative purposes, where there is little wear or abrasion, a thickness of 0·0003 in (0·0076 mm) will give lasting resistance to tarnishing. These thicknesses apply only to deposits on copper and copper-base alloys. When steel is the basis metal, even if a copper undercoat is used, the thickness of the plate should never be less than 0·0005 in (0·013 mm) and for some purposes a thickness of 0·001 in (0·025 mm) may be needed.

The electrolyte for the process has the following optimum composition:

	Gm/litre	Oz/Imp. gal.
Stannous chloride (contains 52·6 per cent Sn)	50	8·0
Nickel chloride (contains 24·7 per cent Ni)	240	38·4
Sodium fluoride	38	6·1
Ammonium bifluoride	17·5	2·8
This make-up gives:		
Total fluorine	29	4·6
Free hydrofluoric acid	6	1·0

The range over which the composition of the electrolyte may be varied in practice is given later. It should be explained that the free hydrofluoric acid is derived from the ammonium bifluoride, which is to be regarded as one molecule of ammonium fluoride associated with one molecule of free acid. In making up the bath it is important to ensure (a) that the chemicals used are of a sufficiently high degree of purity; (b) that the correct method of preparation is employed.

Plating equipment

Since the electrolyte is corrosive, all parts of the plating equipment likely to come in contact with it must be properly protected. Nickel, some plastics, and certain rubbers are not attacked by the solution. Any plastic or rubber must not only be able to resist the solution, but, equally important, it must not contaminate the solution by the leaching out of impurities even in traces. For the latter reason, plastics containing a plasticizer are generally unsuitable. Some plastics that are chemically satisfactory have insufficient mechanical strength at the working temperature of the bath to allow them to be used for tank linings. Several grades of natural and synthetic rubbers have given satisfactory results. With rubber, the important considerations are the free sulphur content and filler content, both of which should be as low as possible. For small installations, Perspex and Polythene are excellent tank linings, but their high cost makes them less attractive for large plants. An alternative that can be confidently recommended for both large and small units is neoprene ebonite.

Except for small units, which may be fabricated from Perspex or Polythene sheet strengthened by external metal bands, or made of moulded plastic, the plating tank should be of steel lined with rubber or plastic. As the electrolyte is operated hot, some means must be provided for heating the solution. An external water jacket, internal electric immersion heaters or steam coils may be used. Internal heaters must be either of solid nickel or of steel that has been very heavily nickel plated, at least 0·002 in (0·05 mm) of nickel. In operation the electrolyte must be maintained at a temperature of 65-70 deg C (149-158 deg F). Thermostatic temperature control is an advantage in large installations.

When hot, the tin-nickel electrolyte may slowly evolve hydrofluoric acid, especially after fluoride additions have been made, and as this acid is toxic, full precautions must be taken to protect the operators from inhaling it. Therefore the plating tank must be provided with means for removing vapour from the surface of the electrolyte or of deflecting it away from the operatives. The best method is by exhaust ducts running along the two longer sides of the tank; the surface of the ducts on which electrolyte might condense and drain back into the bath must be protected by rubber or plastic in the same way as the tank, while the upper parts of the ducts should be protected by acid-proof paint. As the cathode efficiency of the tin-nickel electrolyte approaches 100 per cent there is no evolution of spray from the solution.

Since it is essential that the electrolyte is kept scrupulously clean at all times, filtration is of particular importance. Continuous filtration is advised. Both the filter and the pump must be designed to resist the electrolyte, and the filter should be of a type that permits the use of paper, supported if necessary by cloth, as the filtering medium. Filtering fabrics other than paper are generally unable to retain fine particles which, if not removed, cause pitting. Treatment with activated carbon should be restricted to occasions when the electrolyte has become contaminated with organic matter, since it is very difficult to prevent the passage of fine particles of carbon through the filter into the bath. Any carbon that does enter will cause pitting.

The equipment should include mechanism for reciprocating the work slowly over a few inches. This reduces any tendency to burning on high current density areas and helps to ensure that the plate is consistently bright. Vertical reciprocation is preferred to horizontal reciprocation.

Anodes

For commercial applications at the present time, the simultaneous use of separate nickel and tin anodes is recommended. For ease of control it is an advantage to have separate anode circuits so that the current through the tin and nickel anodes can be adjusted independently. The tin and nickel used

for the anodes should be of the highest purity commercially available. Lead is a most objectionable impurity; it is worth paying a premium to obtain tin anodes that are virtually lead-free. Anode hooks should be of nickel, and all the anodes, both nickel and tin, must be bagged. Terylene cloth or stout nylon may be used for the bags; the fabric used should have a close texture to retain sludge.

If the bath is likely to remain idle for several days, the tin anodes should be withdrawn; the nickel anodes should at all times be left in the electrolyte. Anodes need not be washed on removal; for preference they should be hung up and allowed to drain into the drag-out tank. As the anodes are bagged, it is not easy to see when they are almost consumed, but this condition is indicated by a marked rise in anode voltage.

It is not possible to state precisely the bath voltage, since this depends upon the dimensions of the installation, on the distance between the anodes and cathodes, and on other factors. However, as a general guide it may be taken as in the range 2·0-3·5 volts. For barrel plating a higher voltage is necessary. The tin anodes should operate at about 0·1 volt higher than the nickel anodes.

For the cathode current density, the recommended range is 15-25 amp/ft^2. The cathode efficiency is 100 per cent. The current density on the nickel anodes is not critical and may be anything in the range 10-50 amp/ft^2. On the tin anodes the current density must not be less than 50 amp/ft^2. Lower current densities cause excessive sludging of the tin anodes. Current densities considerably higher than 50 amp/ft^2 are permissible; for example, the figure can be doubled if the plating conditions make it convenient to do so. The anode efficiencies for both nickel and tin approach 100 per cent. In practice, the bath remains in fair balance if the current is equally divided between the tin anodes and the nickel anodes. This current subdivision is recommended. Loss of tin through oxidation should be compensated by the addition of stannous chloride and not by redressing the anode current balance.

The tin-nickel electrolyte is sensitive to metallic and organic impurities. Organic impurities should be completely absent, and the maximum limits for metallic impurities are:

Lead	25 parts per million
Copper	0·2 g per litre
Antimony	0·4 g per litre
Cadmium	1·5 g per litre
Zinc	1·5 g per litre
Iron	0·5 g per litre

High grade chemicals must therefore be used in making up the bath. There is no difficulty in obtaining all the bath ingredients commercially in a sufficiently high state of purity. The Tin Research Institute can advise as to where suitable chemicals can be obtained. Electrolyte preparation is facilitated if two tanks are available so that the solution can be filtered from one into the other. The make-up procedure is:

First the plating tank is filled to two-thirds of its volume with softened or distilled water, and the water temperature is raised to about 65 deg C. The total amounts of ammonium bifluoride and sodium fluoride required for the final volume of the electrolyte are added and dissolved by constant stirring. The stannous chloride is then added and dissolved, followed finally by the nickel chloride. This solution is allowed to stand for several hours and is then filtered, first through activated carbon and then through paper. Filtering through very closely woven paper is essential to remove every trace of carbon, since residual carbon causes serious pitting at the cathode. As a substantial volume increase accompanies the dissolution of the chemicals, the final volume will probably be about correct, but if the volume is low it should be adjusted by the addition of softened or distilled water. The *p*H of the solution should be about 2·5.

Process operation and control

This tin-nickel plating process is simple to operate and easy to control. Success depends mainly upon correctly maintaining the composition of the bath, particularly in regard to the concentrations of stannous tin and fluoride, and on observing the recommendations given for current densities and working temperature. Cleanliness of the electrolyte is of primary importance. Freedom from suspended impurities is essential, and it is equally important that the electrolyte be free from dissolved impurities, many of which, even in relatively small amounts, spoil the deposit.

The most common dissolved impurities are metals, which may enter the bath through careless operation. A frequent cause of this type of contamination is the practice of leaving work to be plated suspended in the bath for some time before the current is switched on. It must be remembered that the electrolyte attacks all common metals and alloys with the exception of nickel, and if, for example, steel or brass articles are left idle in the bath even for a very short time, they will begin to dissolve in the solution.

If the bath is correctly operated, almost as much metal dissolves from the anodes as deposits on the cathode, and the concentrations of tin and nickel in the electrolyte do not alter rapidly, although there is a tendency for the amount of stannous tin in the bath slowly to diminish. There is some loss both of the metals in the bath and of fluoride through drag-out, but most of this can be reclaimed by transferring the work to a drag-out rinse before rinsing proper; the loss from the main tank can be corrected by making up from the drag-out tank, which should be maintained at about *p*H 2·5.

Fluorides are slowly lost while the bath is at working temperature, and as the appearance and particularly the brightness of the plate, depends largely upon the fluorine content, these losses must be made good. Loss of fluoride occurs by precipitation of a compound and by drag-out; the first is the more important. Primarily the production of plate with the correct composition and maximum brightness depends on (a) maintaining the stannous tin content within the limits 22-26 gm per litre (3·5-4·2 oz per Imp gal) and (b) ensuring that at least sufficient fluoride is present to provide four fluorine atoms for every stannous tin atom, and six fluorine atoms for every stannic tin atom in the electrolyte. Compliance with these requirements is of fundamental importance. The aim should be to keep the electrolyte within the following composition range:—

	Gm/ litre	Oz/ Imp gal
Stannous tin	22-26	3·5-4·2
Stannic tin	6-12	1·0-2·0
Nickel	50-60	8·0-9·6
Sodium	15-20	2·4-3·2
Total fluorine	28-38	4·5-6·1
Free hydrofluoric acid	5-6	0·8-1·0

Organic contamination of the bath must be avoided. Such contamination causes pitting of the deposit and in extreme cases may lead to suppression of the nickel with the result that only dull tin is deposited. A solution so contaminated can be rectified by treatment with activated carbon. The best way to carry out this treatment is to pump the solution into a separate tank, stir in about 1 lb of activated carbon per 100 gallons, allow to stand for three to four hours and then pump back into the plating tank through clean filter paper. If the electrolyte has become contaminated with metallic impurities, rectification is best effected by a plating-out process. The bath should be operated overnight on dummy steel cathodes at a current density not exceeding 5 amp/ft^2.

In preparing the work for plating, the usual care should be taken to ensure the complete removal of surface oxide, grease, etc. The tin-nickel deposit has little self-smoothing action and therefore its brightness depends upon the quality of the surface on which it is deposited. If the basis metal is highly polished, the plate will be bright and, in general, may be acceptable as it comes from the bath. On irregularly shaped articles the plate may show a slight lack of brightness on certain areas, but a very slight buffing will bring up the full lustre.

As the tin-nickel electrolyte has exceptionally high throwing power, the use of internal anodes is rarely necessary. When work is racked precautions must, of course, be taken to ensure that one article does not shield another. If there is more than one row of cathodes in the bath, additional rows of anodes must be provided so that there are anodes on each side of the work.

Topping up while the bath is in operation will lead to a cold, dilute

layer in the upper part of the bath, and any work plated in that layer is liable to come out dull. A good way to top up is through the filter. At all times it is particularly important to ensure that the electrolyte is uniform in temperature and concentration throughout its bulk. At a current density of 25 amp/ft^2 a thickness of 0·0005 in is deposited in 15 minutes; as the cathode is always 100 per cent, the time for plating a given thickness at any given current density within the working range can be calculated. Barrel plating does not present any difficulty, and special modifications in electrolyte composition or plating technique are not necessary. Barrel plating is best done in immersed or partly immersed barrels.

The tin-nickel deposit is inherently very resistant to corrosion. In the atmosphere, pore-free deposits remain permanently bright and perform as well as, or better than, chromium on nickel. If the alloy is deposited direct on steel, pores must be absent, as otherwise the steel will be attacked at the pore sites. Since it is difficult with any electro-deposit to ensure complete freedom from pores unless the deposit is very thick, it is recommended that steel should be copper plated before the tin-nickel is deposited. The copper undercoat should be at least 0·0005 in thick.

Investigation has shown that the plate is almost completely immune to attack by condiments, including common salt. It is not attacked by dilute mineral acids; it is attacked by concentrated hydrochloric and sulphuric acids, but only very slightly by concentrated nitric acid. In other corrosive media tin-nickel has proved, with one or two minor exceptions, to be very much more resistant to attack than either tin or nickel. As tin-nickel alloy is an inter-metallic compound it is inherently somewhat brittle, and it is not practicable to carry out much fabrication after plating. It must, however, be emphasized that tin-nickel deposits are free from internal stress and hence there is no danger of spontaneous cracking and flaking away from the basis metal that characterizes some bright deposits that are not stress free.

The process described in these notes was originated and developed in the laboratories of the Tin Research Institute. Patents have been obtained in a number of countries to establish indisputable publication of the process. They will be allowed to lapse at the earliest possible moment, but meanwhile licences to use the process will be granted without fee, since it is not the practice of the Institute to make any charge for the use of its inventions.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Electrical Ignition Equipment

By F. G. Spreadbury, M.Inst.B.E., M.A.I.E.E.

London: CONSTABLE AND CO., LTD., 10-12, Orange Street, W.C.2. 1954. 5½ in × 8¾ in. 227 pp. Price 25s.

The author of this book is a Lecturer in Mathematics at the Regent Street and Northampton Polytechnics and a Lecturer in Electrical Science at the Paddington Technical College. In scope, this, the first edition, includes almost all ignition systems at present in use as well as some details of others that may have wider applications in the future. It deals with construction, design and testing, as well as with engine requirements and conditions relevant to ignition. Therefore, it should interest all who are concerned with electrical ignition equipment and the theory of ignition in automobile and aircraft engines.

The book is written at an intermediate level, and some idea of its scope can be obtained from the items dealt with in the first chapter, which is entitled "Ignition of Explosive Mixtures." This chapter occupies 25 pages, and deals with: Inflammability of explosive mixtures; effect of pressure; ignition temperature; arc ignition; theory of ignition; practical ignition conditions; timing; effect of fuel-air ratio; influence of compression ratio; pre-ignition; detonation; lead influence; plug position and practical ignition systems. There are five more chapters, headed: Sparks, spark gaps and sparking plugs; Ignition coils; Magnetos; Construction and design of ignition systems; Testing ignition equipment.

Energy Transfer in Hot Gases

Washington: NATIONAL BUREAU OF STANDARDS, U.S. Department of Commerce, Washington, 25, D.C. 1954. 6 in × 9¼ in. 126 pp. Price $1.50.

This book is, in fact, the proceedings of the N.B.S. Semi-Centennial Symposium on Energy Transfer in Hot Gases, held on September 17-18, 1951. The problems of radiation from flames and hot gases, and basic physical and chemical mechanisms governing energy transfer in those media are subjects of current interest that several divisions of the National Bureau of Standards have been actively investigating. These problems are of both theoretical and practical importance in a number of fields, since improvements in heating devices of all kinds, gas turbines and internal combustion engines depend upon a full understanding of the detailed mechanisms of energy transfer and the combustion process.

Each of the sixteen chapters is a paper read at the Symposium. Some idea of the nature of the work can be gained from the following three headings of papers chosen at random from those in the book. One is entitled "Processes of electronic excitation in relation to flame spectra," by A. G. Gaydon. Another is "Spectroscopic studies of low-pressure combustion flames," by S. S. Penner, M. Gilbert and D. Weber, and the third is "Infra-red emissivity of diatomic gases," by S. S. Penner. Most of the papers contain half-tone and line illustrations.

Strength of Materials

By A. Morley, O.B.E., D.Sc., Hon. M.I.Mech.E.

London: LONGMANS, GREEN AND CO., LTD., 6 and 7, Clifford Street, W.1. 1952. 5½ in × 8¾ in. 583 pp. Price 25s.

Since this book was first published in 1908, it has been widely used not only by engineering students, for whom it was written, but also as a work of reference by practising engineers. When the ninth edition was produced, the need for resetting the type gave an opportunity for complete revision, and some of the chapters and articles were re-written. Notable instances are the work on fatigue, criteria of elastic strength, creep, metallurgical developments of ferrous metals and methods of testing. More use, too, was made of electrical strain energy and the theorems related to it for the determination of elastic deformations. In this, the tenth edition, many small changes and additions have been made, particularly in relation to helical springs, strain gauges and strain analysis. The book is so well known that it is unnecessary to give further details.

AUTOMATION

Recent Developments at Ford Motor Company, U.S.A.

H. W. Perry

COINED by D. S. Harder, vice-president in charge of manufacturing at the Ford Motor Company, automation is the term applied to electrically controlled automatic movement of parts to be processed through and between a connected series of machines which perform many different operations and their unloading at the other end of the line, all without manual handling. When such a production line has been designed, equipped and installed with maximum exactness, the following results are obtained:—

Reduction of floor area requirements in the plant;
Increase in production rate;
Lessened overhead costs;
Closer dimension tolerances;
More precise inspection after operations;
Lower unit cost of products;
Cleaner plant floor, and
Easier, safer, more interesting work.

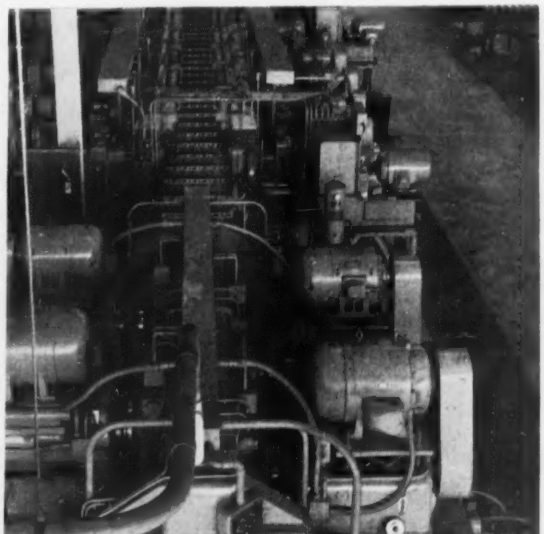
The most notable example of automation in the industry is the new machine line in the Ford Company's Cleveland, Ohio, engine plant which finish-machines rough cast six-cylinder engine blocks, which are produced in the foundry by the new method. Eight-cylinder blocks for Mercury cars are also machined automatically. Machining of the latter is done by a series of forty-two machines connected mechanically and electrically in a serpentine line that occupies more than an acre of floor area and would, if it were straight, extend more than a third of a mile. As the engine blocks move from one end of the line to the other, the machines perform 530 cutting and drilling operations on each casting. Typical machines and a transfer mechanism are shown in the accompanying illustrations.

Development of this system and setting up the equipment required more than six years of engineering study, planning and designing and involved great expenditure for machines, control equipment and setting-up labour. It included provision of equipment for casting and machining engine blocks and heads, machining crankshafts, connecting rods, aluminium pistons and other parts, and the assembling of engines ready for operation. The many factors that need consideration and exacting study in the establishment of automation in a factory are discussed at length in a paper by G. G. Murie, supervisor of design and manufacturing engineering at the Ford engine and foundry division, which was presented at the November, 1953, meeting of a branch of the National Metal Trades Association.

Mr. Murie stated at the start that reduction of costs is the underlying purpose served by the evolution in manufacturing methods in the automobile industry and that the keynote of all planning is proper and simple design of product items for economical manufacture and reduction of production-line maintenance. The extent to which it was possible to introduce automatic production was decided largely upon engineering studies, experience of the automobile company and the machine-building industry, and study of the economics of the subject.

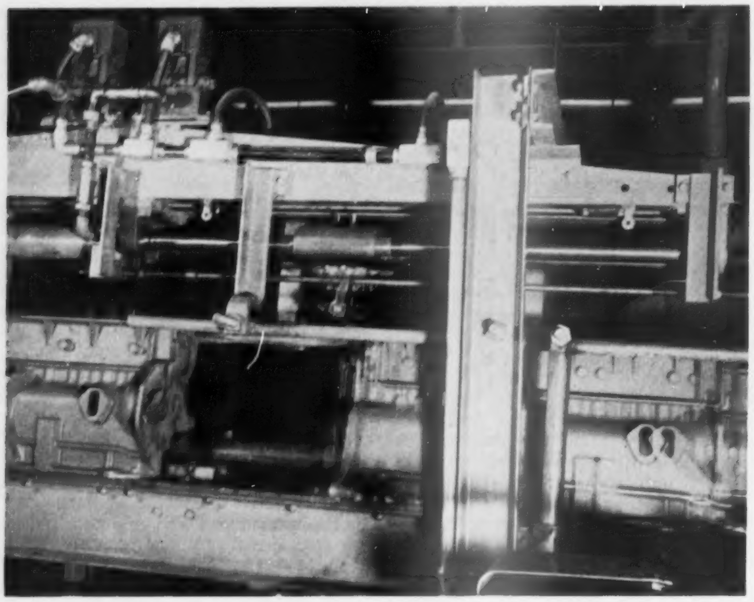
The principal matters requiring solution were: type of machinery needed for transferring parts in process from one machine to the next; use of company-owned standard machines or need of special machines to be built; plant layout of the production line to make best use of the floor area; provision of means for minimizing maintenance to prevent or reduce the time of interruptions of machining operations; automatic disposal of scrap; reassignment of workers so as to utilize mental ability instead of muscular strength; and co-ordination and support of the programme by company management to obtain maximum benefits from the technical skills.

Production of finished engine blocks or engine parts by automatic machines connected in line requires transfer mechanisms which move the units from one machine to the next, places them in the loading stations in correct position for machining and may also then rotate or tilt them for proper loading in the next boring, drilling or cutting machine. Usually a group of devices is needed if the number of machines required for one type of operation differs from that used for the following operation. To synchronize the movements, the loading and unloading of individual machines are interlocked electrically through a control panel so that they do not occur while the machines are going through their operating cycles.



Transfer machines for six-cylinder engine blocks. The five-station machine in the foreground mills both ends

Engine blocks and heads are seldom touched by workmen before final assembly. Instead, electric "nerve centres" direct all movements through the machine line by means of mechanical arms and fingers. Electrical switches send hundreds of information items continuously from a single machine to such a centre. One information post may report that a certain machine is ready to go to work, another post that an engine block on the conveyor is waiting for processing, and a third post informs that steel arms and fingers are poised to transfer the block into working position in the machine. When all these conditions are right, the electric brain turns on the power that actuates the sequence of movements. After operations by one machine are completed, the transfer mechanism shifts the engine block to the next machine, loads it in working position and gets set to repeat the operation with the next block. The brain can direct such operations simultaneously for the



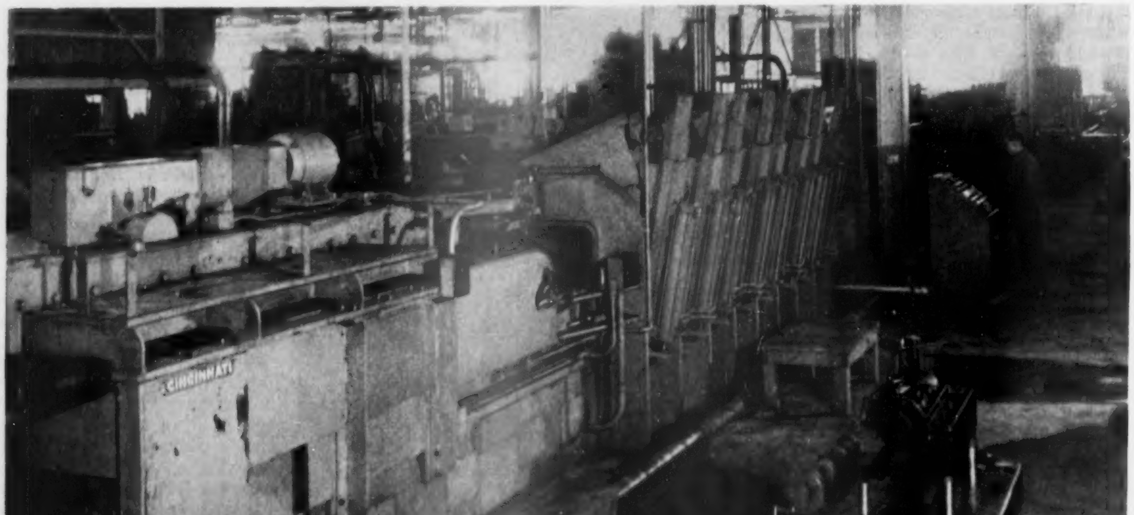
One of the transfer mechanisms for moving blocks between machines

many machines in the line.

Even collection and removal of all swarf from the machines is mechanized and automatic. Covered trenches and troughs in the plant floor extend to all machines that generate more than 40 lb of swarf per day. Three chip-handling methods are used. Chips from dry machining operations on cast iron are moved in a trough which passes through all the machines and is oscillated in a trench by a small motor having an eccentric drive which thrusts the chips along by vibrator conveyor that dumps them into a hopper located in a pit 10 ft square. Interruption of an electric-eye beam when the hopper is filled stops the oscillating conveyor long enough for the hopper to rise automatically and discharge the chips into four-wheeled cars to be hauled to a briquetting house to be processed for re-use in the foundry.

Steel chips from wet machining of crankshafts and connecting rods in a separate automatic production line are removed by an agitator conveyor containing a soluble oil that flushes them down inclined trenches into tanks under the floor. The soluble oil is then pumped back into the system through overhead piping and the chips are scraped by a moving bar from the bottom of the tank and along inclined conveyors to removal carts. Grindings are simply flushed through trenches into receiving tanks. There are sixteen collecting tanks in the wet system, the largest having a capacity of 10,000 U.S. gallons.

Although assembling of the many parts with the finished engine blocks must be done manually, the heavy handling and danger involved in usual methods have been eliminated and the plant floor cleared of conventional assembly-line conveyors. Mechanical arms suspended from an overhead monorail system lift the 180 lb engine blocks as they reach the end of the machining line and carry them automatically to various assembling stations



This machine automatically broaches the main bearing journals and locks, and the top and bottom faces of the cylinder block. Output is at the rate of 150 components per hour

to which they are sent by switches in the monorail lines. The turn of a crank on a lifting arm tilts a block to any position or angle that is most convenient for fitting any of the many parts, such as shafts, connecting rods, pistons, head, valves, carburettor and spark plugs. Any engine that may need more than normal assembly work can be shifted to a side track by merely flicking a selector button, thus avoiding interruption of other assembling operations. There are no hanging chain hoists or other manually operated devices that would require physical effort and distract attention from precision assembling work.

Automation has some disadvantages as well as advantages, as Mr. Murie pointed out. It creates maintenance problems because the entire system is geared to production at a uniform rate per hour, with no possibility of making up for loss of output caused by the breakdown of even a small device or replacement of a cutting tool that fails. Therefore, plans for preventive maintenance were started by Ford executives during the early stages of the new programme. They include the keeping of accurate records of lubrication and expected durability of parts that wear fast or may break and their replacement before they are likely to fail. Standard cutting tools are used wherever possible so that tool change time is reduced to a minimum.

A cardinal innovation in this connection was the development of a tool-control board on which each tool has a definite location and is pre-set to specific standards in readiness for immediate replacement of a worn tool without need of adjustments on the machine. A tool meter on the board is connected to each tool in a machine and pointers on the dial are set to a predetermined number of work pieces to be machined before regrinding of a tool becomes necessary. When any tool in the machine has done its work on that number, the control board shuts down that machine and the tool is changed. The set-up man can determine by scanning the board which tool or tools should be changed at once, and he selects at the same time other tools to change as pointers over a red area on the dial indicate that those in use are close to their limit of working capacity or efficiency. This tool-change procedure effects large financial savings by minimizing tool breakage, the number of production parts damaged in process, and down time of the whole production line. A preliminary phase of preventive maintenance was the simplifying of design details of the system and making many of the devices with a large factor of safety against failure.

Another disadvantage of automation is that engineering costs are very high because so many factors are involved and require most careful planning and hours of designing before starting construction. Then installation of a production system usually necessitates the relocation of machines, which may be expensive and seriously delay production of parts and final assembly of a company's product. However, loss of output may be avoided or greatly reduced by installing the automation line when production is shut down owing to changes in the product model and in machining operations which make rearrangement of machines necessary in any case.

A further disadvantage is that machine tools built by various machine companies differ greatly in design of the loading stations. Careful study and planning of transfer devices must be done if such different machines are to be coupled in a continuous automatic line.

Whether or not it is advisable for a company to adopt an automation system is, therefore, a matter of economics. Keen competition within the industry dictates reduction of costs and faster volume output, but expenditures for machinery and equipment must not be so large as to nullify the objective of financial gain and other benefits. It is important to give careful attention to the question of production equipment amortization. Some equipment may become useless after a year of use owing to product changes or obsolescence. But annual changes in engines and chassis parts may be such that the basic automation system can be used for several years without much additional investment in the machines. In many cases, it was emphasized, correctly designed equipment costs little more than standard devices. All the machinery and equipment in the cylinder-block department of the Ford engine plant in Cleveland is new and of latest design, including many transfer-type machines.

WORM GEARING LUBRICATION

The oil used to lubricate a pair of worm gears has to serve two purposes. First, it serves to reduce friction and wear between the surfaces of the worm threads and wheel teeth; secondly, it helps to carry away the heat generated in the zone of contact. It is known that a very high local temperature occurs at the point of contact between gear teeth, and it is desirable, particularly with worm gears, that there should be a flood of lubricant to prevent excessive heat build-up on the worm threads.

The normal and most widely used method of lubricating worm gears is by means of an oil bath. If the worm is below the wheel, the bath is usually arranged with a level such that the lower part of the worm dips in the bath. If the worm is overhead, it is usual to provide an oil bath in which the lower teeth of the wheel dip.

Oil baths give satisfactory results in all cases except where the speed of the wormshaft is very high. If the pitch line speed of the worm exceeds about 2,500 ft. per minute, there is a tendency for centrifugal action to throw the lubricant off the teeth and threads. It is then desirable to use a jet of oil under pressure and directed on to the entering side of the worm threads and wheel teeth. If the mass of the oil bath and gear case is not sufficient to keep the general temperature of the oil within reasonable limits, it may be desirable to circulate the oil through a cooler.

Lubrication by oil mist is not generally satisfactory, because, although it may provide a film of oil between the teeth and threads, it does not usually provide the flood of oil necessary to carry away the heat. For the same reason, grease lubrication is generally unsatisfactory for worm gearing unless the speed is very slow. The gears tend to cut a channel through the grease, so that in time there is no lubricating film and no flow of lubricant to carry heat away.

The British Standard Specification for worm gearing suggests that the temperature rise of the lubricant should not exceed 100 deg F above the surrounding atmospheric temperature. Without using a thermometer, it is possible to obtain a rough guide to the temperature of the oil bath on a running gear by applying the fingers or the palm of the hand to the outside surface of the gear case. At a temperature of 140 deg F in the oil bath, the hand can be maintained permanently on the outer surface of the box, but at a temperature of 160 deg F, it is usually only possible to maintain contact for about five seconds.

The viscosity of the oil for worm gears depends upon the size and speed of the gears and on the conditions under which they are running. John Holroyd and Co. Ltd. recommend oils varying from S.A.E.140 to a lighter oil of about S.A.E.50. Cold starting conditions require a lubricant that will flow readily at low temperatures, whereas a gear working in a hot atmosphere generally calls for a heavy grade of oil. There is still much room for further research on the subject, but John Holroyd and Co. Ltd. consider that the best results are obtained by using as heavy an oil as circumstances will permit. For worm gearing they have never found any advantage in the use of extreme pressure lubricants or even in the use of mild additives.

FAST-ROLL SPOT WELDING

A New American Technique for High Speed Operation

BY means of a new technique and modified equipment, the Ryan Aeronautical Company of America have recently increased welding speeds by more than 100 per cent and at the same time improved the quality of the welds. Specially equipped Taylor - Winfield machines are employed. They produce gas-tight seam welds at 26 in per minute in comparison with speeds of between 6 and 12 in per minute that were obtained with the equipment hitherto used. Spot welds on 1 in spacings can be effected at 68 in per minute in contrast with the former rate of 21 in per minute. This improved performance is obtained in welding tough stainless steels, Inconel X and W, Haynes Stellite No. 25 and other crack-sensitive alloys used in the fabrication of after-burners, variable nozzles, exhaust cones, and other high temperature jet engine components.

Spot welding machine with Taylor-Winfield control

These new Taylor-Winfield machines have been equipped with fast-indexing mechanisms for roll-spot welding. The devices are air-operated, electronically - controlled units consisting of two horizontally-opposed bellows that actuate an over-riding clutch. The clutch, which is actuated by standard air-line pressure at 80 lb/in^2, rotates the upper wheel electrode of the machine. When a switch is pressed, the continuous electric seam welding drive mechanism of the Taylor-Winfield machine is detached and the roll-spot drive takes over. It can be set by indicator to index in single or multiple movement and to spot weld at spacings from zero to five inches.

The new controls are a distinct advance over the Geneva-type used previously. With the Geneva drive only limited flexibility is available, since the proportion between the dwell time and the indexing time is a fixed ratio that depends upon the gearing and the drive mechanism. It can be varied only in fixed increments. The Taylor-Winfield drive provides infinitely variable dwell and indexing time ratios.

Originally the new mechanism was developed to effect spaced out spot welds at increased speeds, but another application—faster seam welding—has also been conceived for it. With continuous drive machines, certified gas-tight seam welds could not be made at speeds exceeding 6 to 12 in per minute in the crack-sensitive alloys. When these rates were exceeded, the metal temperature rose to the point where undue shrinkage and distortion occurred and cracks developed.

Careful analysis of the phenomenon disclosed that with continuously rotating electrodes, a large number of overlapping spot welds was necessary to produce a gas-tight seam, because each spot weld was inferior to spot welds produced under static conditions. Consequently, the Ryan technicians directed their efforts towards the speediest intermittent operation commensurate with high quality.

The Taylor-Winfield drive was speeded up beyond its already rapid action. Exhaust mufflers were changed and a design that exhibited less back pressure was installed. In addition, an improved air-intake valve with a faster action was incorporated and a new type of faster-meshing gearing was developed to provide greater speed.

As a result of these modifications, it was possible to employ completely new performance ratios. Dwell time, during which the electrodes are motionless, was reduced to the shortest possible interval compatible with the production of a sound weld structure. Indexing time, which is the time lost in making the switch from one spot weld location to the next, was also greatly reduced. Owing to the flexibility of the equipment, these time ratios can be co-ordinated to give the optimum values for different materials and thicknesses.

When precision control of intermittent operation was effected it was found that better spot welds were being produced than under continuous operation. The spot weld nuggets were sounder because the metal was held under forging pressure during the cooling cycle and a larger, stronger nugget was made. Furthermore, as the sheets were held during cooling, shrinkage and distortion from heat were greatly reduced.

The fact that sounder spot welds could be produced by the new technique suggested that gas-tight seam welds could be formed with fewer spots and at greater speeds. This has proved to be correct. For example, on a typical assembly formed of two sheets of 0·045 in Inconel W, it had previously been necessary to make 11 overlapping spot welds per running inch to produce certified seams under continuous drive, and because of the proximity of the welds and the consequent accumulation of heat, 12 in per minute was the maximum speed that could be employed. With the new roll-spot technique, certified seams can be made with only 5½ spot welds per inch and 26 in of seam can be produced per minute.

The new technique is equally satisfactory for spaced out spot welding. For example, in spot welding a General Electric J-47 jet engine tail pipe, 300 spot welds are made at 1 in centres on one seam. Welding is effected at the rate of 68 in of seam per minute against the rate of 21 in per minute with the older methods. Improvements are now being developed to allow speeds of 80 in per minute to be employed. The development is economically sound. A Taylor-Winfield accessory drive costs about 15 per cent of the price of a new machine. For this outlay, the machine capacity is at least doubled.

Although this, like other welding developments made by the Ryan Aeronautical Company, is intended primarily to meet the exacting requirements for aircraft jet engine components in difficult materials, it can also have other applications.

ROLLING MILLS

The New Plant of T.I. Aluminium Ltd.

DURING the past few years there have been great advances in this country in the manufacturing techniques used for the production of rolled aluminium products. These developments have entailed very large capital investment, and have been undertaken with two objects in mind—first, the improvement of the product and second, to reduce the manufacturing costs and the selling prices of aluminium mill products. The first aim has been fully achieved, and present-day products have much greater uniformity of physical and mechanical properties than comparable pre-war products. At the same time, the new techniques also give a better degree of surface finish and greater dimensional accuracy on plates, sheets and strip.

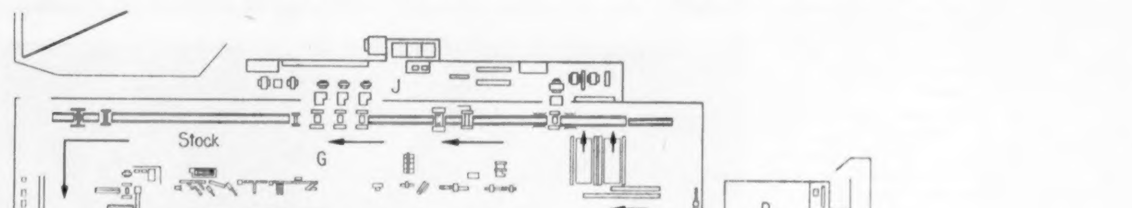
With regard to the second aim, the question of price, the position is not so favourable, for two reasons, both outside the control of the organizations comprising the British aluminium industry. The improved manufacturing techniques now employed have considerably reduced actual production costs, but unfortunately, the price of aluminium, which for all practical purposes is established by the producers in Canada, is to-day far higher than it was a few years ago.

The price of aluminium pig used in this country is virtually controlled by the large Canadian producer, and there is reason for thinking that this near monopoly leads to higher prices than actual pig production costs warrant. However, a more serious factor so far as this country is concerned was the devaluation of the pound in relation to the dollar, since in terms of British currency this had the effect of raising the price of aluminium pig proportionally. In fact, before devaluation, the price was in the order of £96 per ton, and with pig at this price, the industry in this country justifiably considered that sheet and strip could be produced and sold at prices that would encourage engineers to take advantage of the special characteristics of aluminium and aluminium alloys. Since devaluation, however, the price of aluminium has been in the order of £156 per ton, and for purely economic reasons many potential users have continued to use cheaper steel products, although for functional reasons aluminium products might well be preferred.

British aluminium fabricators can do nothing regarding the price of the material from which their products are produced, but they have done much to improve the quality of the products, and at the same time keep prices as low as possible, by employing improved manufacturing techniques. A splendid example of what is being done by the industry is afforded by the new fabrication plant of Tube Investments Ltd. at Resolven, near Neath, South Wales, which is the subject of these notes.

Tube Investments Ltd. made their first incursion into aluminium fabrication in 1932 with the production of aluminium tubing, quickly followed by the production of extruded sections. Since then, their interests in the industry have been greatly extended. In 1946 a detailed survey was made of probable future demands for rolled aluminium products, and at the same time, the most advanced rolling techniques in the U.S.A. and Canada were studied. From the survey, it was estimated that the greatest potential demand would be for mass-produced standard sheet and coiled strip in commercially pure aluminium and in low and medium, non-heat-treated alloys, to close dimensional tolerances, with good deep drawing and forming properties, and having a clean, regular surface finish. The Resolven plant has been planned specifically to meet this demand.

The layout of the plant is shown in an accompanying illustration. Essentially, the production may be regarded as comprising three departments: a foundry, a slab preparation shop and a rolling mill department, with this last department having a hot rolling line and finishing lines. The whole plant is laid out to give flow production from the delivery of ingots to the foundry to the delivery of finished materials to the inspection department for examination before despatch. As much use as possible is made of mechanical handling equipment.



Site plan of T. I. Aluminium Ltd. rolling mill at Resolven

A. Re-melt department B. Laboratory C. General stores D. Quantometer E. Foundry F. Slab preparation shop G. Hot rolling mill line H. Cold finishing line J. Hot mill motor house K. Cold mill motor house L. Works offices M. Sheet rolling annexe N. Future extension O. Inspection and despatch P. Main sub-station Q. Main weighbridge

The arrowed lines indicate material flow from ingot to finished product

Charging a melting furnace

There are six reverberatory gas-fired Gibbons melting furnaces in the foundry. They are arranged in pairs with a semi-continuous casting machine of the Company's design between each pair. Each furnace has an integral holding bath to which the molten metal from the melting hearth is transferred by gravity. In two pairs of furnaces, the melting hearth holds 13 tons and the holding bath 7 tons, while in the latest the melting hearth and the holding bath each holds 10 tons. Before the holding bath is tapped for casting, a sample is taken and is checked for chemical analysis on a Quantometer in a matter of seconds. The Quantometer has completely eliminated the need for lengthy routine chemical analysis by a relatively large staff of laboratory assistants, but, even more important, it gives complete assurance that the chemical composition of the metal in the holding bath is in accordance with specification before the slab is cast.

In casting, molten metal runs by gravity direct from the holding bath down launders, in which are incorporated automatic flow valves, into the head of the casting machine. The number of slabs cast simultaneously depends upon the slab size. For example, the largest slabs, which are 50 in by 10 in cross section by 13 ft long, are cast two at a time. Smaller slabs may be cast four at a time. Two factors must be closely controlled during this semi-continuous casting process, the rate of pour and the rate of drop. The first is governed by the automatic flow control valves incorporated in the launders leading to the head of the casting machine, while the ram drop is controlled by means of a Weatherley infinitely variable hydraulic pump unit.

In addition to the equipment for casting slabs for subsequent processing in the rolling mills, the foundry also includes equipment for casting extrusion ingots for subsequent processing at another of the Company's factories. These are also produced by semi-continuous casting. An ingot cast by the semi-continuous process has more even grain size and is more homogeneous than one cast into a fixed mould, but to ensure that only 100 per cent material is supplied for extrusion, every ingot is tested on ultrasonic equipment for freedom from internal discontinuities.

From the foundry the slabs are transferred to the slab preparation shop. There they are first sawn into suitable lengths for rolling on Noble and Lund Fluidfeed sawing machines with 38 in diameter blades. Sawn lengths are transferred to John Holroyd and Co. Ltd. scalping machines to have the top and bottom faces machined to remove all oxides and surface irregularities. The scalping machines are, in effect, vertical spindle millers with 51 in dynamically balanced cutters running at a peripheral speed of 6,000 ft per minute. These machines are arranged in pairs with a turnover gear between each pair, so that one side is machined on the first machine of a pair, the slab is turned over, and the other side is machined on the second scalper of the pair. After being cut and scalped to standard sizes, the slabs are held in stock against rolling requisitions.

As required for rolling, the slabs are conveyed by crane from the slab preparation shop to a slab conveyor which carries them to the rear end of two double-chamber Gibbons pre-heating furnaces. The internal dimensions of the heating chamber are 9 ft wide by 5 ft 3 in high by 63 ft long. Heating is by gas-fired radiant tubes, and forced-air circulation is employed in nine zones, each with individual control. Charging is effected by means of overhead hoists that up-end the slabs and place them on chairs for traverse through the pre-heating furnaces. At the discharge end of the furnaces a hydraulically-operated gear deposits the slab directly on to the live roller feed table of the hot mill.

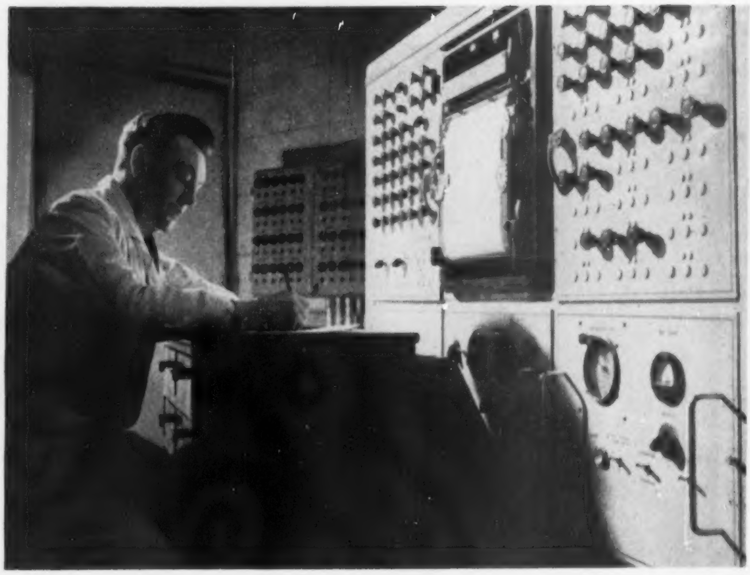
The breaking down mill, the first in the hot mill line, was built by the Brightside Foundry and Engineering Co. Ltd. It has rolls 35 in in diameter × 80 in long, running in fabric bearings. This mill is directly driven by a 1,800 h.p., 40/52 r.p.m. D.C. reversing motor, with a maximum

A semi-continuous casting machine. Three slabs are being cast simultaneously

operating horse-power of 4,950 supplied from a motor generator set driven by a 2,500 h.p. 6,600 volt induction motor and a 65,000 h.p. second flywheel. The drive is arranged to give rolling speeds up to 475 ft per minute.

Once the pre-heated slab has been discharged on to the live roller feed table, which is 138 ft long, control of the breakdown rolling is effected by one man seated in the "pulpit." The maximum weight of slab is 3,500 lb and in 11 passes through the 2-high reversing hot mill the thickness is broken down to 0·25 in, that is to $\frac{1}{40}$ of the original thickness. The total length of the run-out table from the breaking down mill to the first of the 3-stand 4-high tandem mills on which the hot rolling is finished is 500 ft, divided into two individually-controlled sections. A pair of side manipulators is provided on either side of the breakdown mill and a further pair adjacent to the tandem mills.

Between the breakdown mill and the tandem mills the following equipment is mounted to bridge the run-out table, and in this order: a 1 in × 98 in up-cut shear by Head Wrightson and Co. Ltd., an edge trimmer with 21 in diameter cutters adjustable to 70 in width by Loewy Engineering Co. Ltd. and a Head Wrightson $\frac{1}{8}$ in × 54 in up-cut shear.



The Quantometer

From the final end trimmer, the part-rolled material passes on to the last of the individual runs of the run-out table to be fed to the tandem mills. These mills by W. H. Robertson and Co. Ltd. have work rolls 17 in diameter × 60 in long and 42 in diameter backing rolls running in roller bearings. Each stand is driven by a 1,500 h.p. D.C. motor, and the drives are arranged to give rolling speeds up to 200 ft per minute on the first stand, 420 ft per minute on the second stand and 500 ft on the third stand. Material up to $\frac{1}{8}$ in thick in coils 34 in diameter × 50 in wide is coiled on a drum coiler at the exit end of the mill, while thicker material is coiled on a three-roll coiler. Coils are run by gravity conveyor to a weighing machine and thence to a cooling conveyor. During the passage of the material through the three tandem mills, special Metropolitan-Vickers exciter sets control strip tension and inter-stand speed.

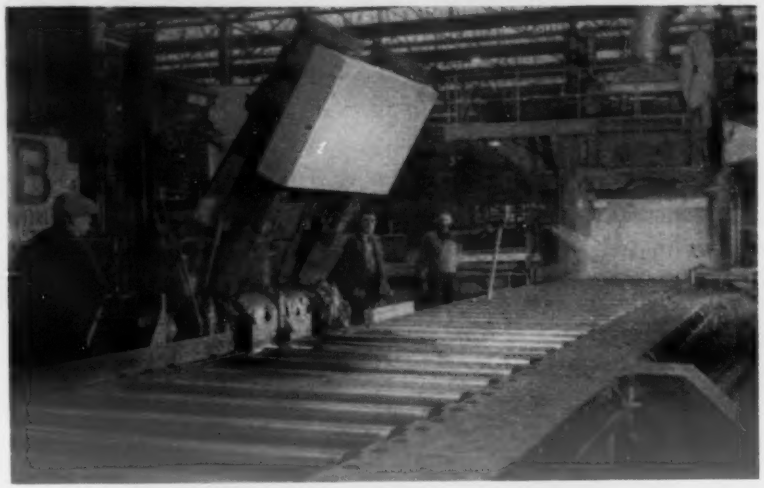
Coolant plays an important part in the successful operation of this hot mill line. The soluble oil employed is carried in a 50,000 gallon storage tank equipped with electric immersion heaters to maintain the oil at a constant temperature, particularly at week-end shut-down. A cooling tower removes any excessive heat. Circulation is effected by two 120 h.p. Pulsometer centrifugal pumps. They deliver oil to the mills at the rate of 2,000 gallons per minute at a pressure of 60 lb/in^2. From the mills, the soluble oil drains into glazed cooling and filtering tanks, whence it is pumped to the cooling tower by two Sulzer vertical spindle pumps.

Scalping a rolling slab ready for hot rolling

Here it may be as well to summarize the products of the hot mill line. In general, this line is used to process cast slabs 10 in thick and weighing up to 3,500 lb into continuous coil with a minimum thickness of 0·062 in. In physical properties the strip is equivalent to hard temper in commercially pure aluminium and non-heat-treatable alloys. These materials can be supplied in coils up to 4 ft wide or as flat sheet up to 4 ft wide × 20 ft long. The surface finish obtained on this hot mill line is sufficiently good to make the product suitable for use in many applications, without any further processing other than a final trimming. Of course many applications require a higher quality of surface finish, and for such applications the products of the hot line are transferred to one of the cold finishing strip mills for further processing.

Sheared blank in the heat-treatable alloys for subsequent cold rolling is another product of the hot mill line. Owing to the power and the advanced rolling techniques employed in this plant for the production of hot rolled strip and sheet, considerably less cold rolling is necessary than would be required for hot rolled strip produced by general rolling methods. This results in an improved product with less directionality.

Coils for cold rolling are first

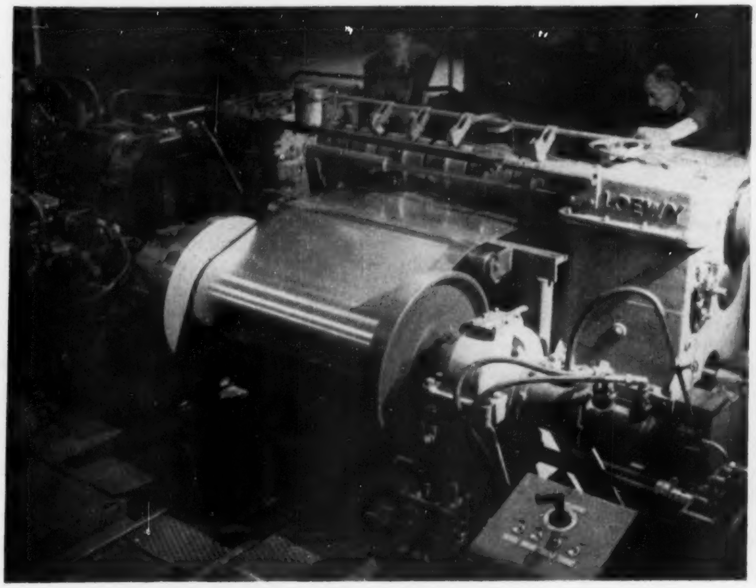


Transferring a slab from the pre-heat furnace to the roller feed table for the hot rolling line

The run-out side of the breaking-down mill through which the slab makes 11 passes

Part of the hot mill line with the three tandem mills in the background

Electric furnaces for annealing hot rolled strip before cold rolling



Loewy hydraulically actuated machine for edge trimming and slitting coils

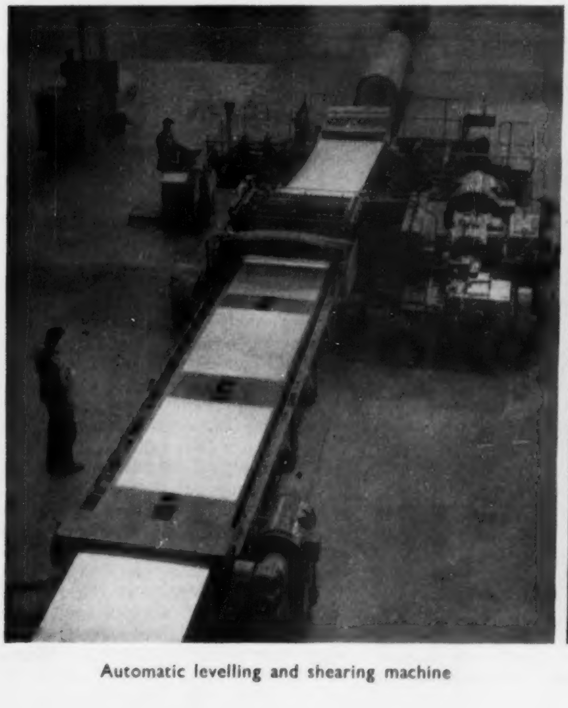
annealed. They are transferred from the hot mill cooling conveyor to a charging machine, with an up-ender and loading crane, that is arranged to charge work into any one of three electric annealing furnaces. External heaters are employed in these furnaces and air is continually circulated at high speed between the heaters and the furnace. The heater unit is sub-divided to give 140, 215 and 280 kW, or any combination of these ratings. Each furnace will anneal two tons of coiled material per hour. The furnaces and the charging machine are by Stordy Engineering Co. Ltd.

For cold rolling the two main units are 17 in × 42 in × 60 in 4-high non-reversing strip mills. Each is equipped with in-going conveyors and roller bridles on the one side and with belt wrappers and drum coilers on the out-going side. One mill was supplied complete by W. H. A. Robertson and Co. Ltd., who were also responsible for all strip rolling auxiliary equipment on the other although the mill itself was supplied by Davey-United Engineering Co. Ltd. Each of these mills is driven by a 1,500 h.p. D.C. motor and the maximum rolling speed is 1,000 ft per minute. Constant tension over a wide range of tensions and build-up is maintained through special exciter sets. A typical cold rolling reduction on coils 50 in wide and weighing 3,000 lb is 0·030 to 0·014 in.

Flood lubrication is provided at each of the cold rolling mills and each has a most effective fume extraction plant. Oil is supplied to the mills from a single storage tank. Harland Duoglide pumps deliver oil to the mills, and similar pumps take the oil from the draining tanks and deliver it to a battery of Lilos twin cartridge strainers, each of 3,000 gallons capacity. From the strainers, the oil passes to a 10,000 gallon tank, whence it is passed through four Stellar filters and thence to the storage tank for re-circulation.

There is a considerable demand for coil strip, foil stock that calls for further cold rolling. For this product, cold rolled coils are first annealed and then temper rolled on a 2-high non-reversing strip mill by W. H. A. Robertson and Co. Ltd. This mill is designed to reduce strip from 1 mm to 0·30 mm in thickness. It incorporates several novel features, including hydraulically-controlled back tension equipment. Both the de-coiler and the coiler are arranged to provide variable tension. This mill has an oil circulation and cleaning plant similar to, but on a smaller scale than, the plant for the two main cold rolling units.

To suit customers' requirements cold rolled coils have to be supplied in widths that differ from the rolling width. For edge trimming and slitting the coil, a special machine, developed



Automatic levelling and shearing machine

Roll forming machine for corrugated sheet

and built by The Loewy Engineering Co. Ltd., has been installed. The machine is driven by Vickers V.S.G. hydraulic pumps and motors. The recoiling train operates at speeds up to 500 ft per minute and can maintain a tension of 5,000 lb on the strip. Both the speed and the back tension are infinitely variable. Very accurate control of these important factors is obtained by using variable output hydraulic pumps directly coupled to hydraulic motors.

Here it may be remarked that in the U.S.A. it is common practice for aluminium sheet to be supplied in the form of coils, a practice that can lead to substantial production economies. In this country this practice is not nearly so common, but it is encouraging to note that the use of coiled aluminium sheet, in place of flattened sheet, is growing.

Even though the use of coiled sheet becomes more general practice, there will always be a demand for flattened sheet, and these mills are equipped for turning out high quality flattened sheet. There are two lines equipped for the continuous levelling and shearing of sheet from coil. One for material up to 48 in width and the other for material up to 54 in width; each line can produce sheets from 2 ft 6 in to 20 ft long, and each can handle material at speeds up to 300 linear feet per minute. All the equipment for these lines was supplied by W. H. A. Robertson and Co. Ltd. At the end of each line there is a belt conveyor to transfer cut and flattened sheet to a stacking frame.

A roll forming machine built to the Company's design is installed in line with the 54 in levelling and shearing machine, to which it is connected by a removable length of belt conveyor. This machine is used for the production of standard corrugated sheet. For certain applications an even higher standard of flatness is required than the good standard obtained from the continuous machine. Roller levellers and hydraulic stretching machines have been installed for producing sheets with the best possible degree of flatness.

A fully automatic shearing line has been installed for accurate side and end trimming on sheets that have been roller or stretcher levelled. This comprises four guillotine shears supplied by Head Wrightson and Co. Ltd. One of the guillotines for side trimming and one for end trimming can be adjusted in relation to their counterparts for width and length adjustments. Sheets are fed through the unit on a belt conveyor. This plant incorporates provision for the mechanical removal of the scrap edges In the same section of the mill there are several auto-feed power presses for the production of a wide range of circular blanks; there are also circle cutting machines for sizes that are too large for production on the power presses.

Much of the production from the automatic shearing line is intended for use in deep drawing operations, for which the temper of the material as it leaves the guillotine is not wholly suitable. Such material is, therefore, given a flash annealing operation in a special Stordy continuous annealing furnace. This furnace is approximately 100 ft. long. It comprises a heating chamber into which air, pre-heated by means of electrical resistance external heating elements, is circulated through the chamber. Sheets, circles, or blanks are conveyed through the furnace automatically at a pre-set speed on a series of heat resisting tapes supported on rollers. From the heating chamber, the work passes into a cooling chamber from which it emerges ready for stacking.

In view of developments such as those described in these notes, it is interesting to speculate as to whether the automobile industry in this country is likely to offer a wider market in the future for rolled aluminium products. At present, fairly considerable use is made of aluminium sheet for bus and coach bodies, but for passenger cars its use is limited to the higher price range and sports models. That rolled low alloy aluminium sheet is a technically practical material for a passenger car body is now an accepted fact; the work carried out by Panhard in France has proved this conclusively. It is also accepted that aluminium, apart from weight reduction, has certain properties that make it a better material than sheet steel for bodywork. The most important of these is corrosion resistance. However, because of the price differential there is little prospect that aluminium will replace steel as the chief body material, but it is arguable that the difference in end cost would be much less than the difference in the prices for sheet steel and sheet aluminium would suggest. An exhaustive investigation into total comparative costs might well produce conclusions that would controvert accepted views on the subject.

ROLLER BEARING LUBRICATION

Problems and Solutions

AN article entitled "Some Lubrication Problems with Roller Bearings and Their Solution," by R. H. Dubois, has been published in *Lubrication Engineering*, February, 1954. In it, the author states that tapered roller bearings in cars and trucks are almost entirely grease lubricated. Light greases are normally used, but operators of larger trucks may use a heavier grease to minimize leakage through oil seals. Under severe conditions, still heavier greases are used, but may "channel." The employment of greases of greater fluidity is recommended.

Spalling

Spalling in truck wheel bearings has been traced to etching of races caused by moisture entering the bearings through the oil seals. Visual detection of impending failures, and spalled, etched or discoloured bearing replacement, is easy with proper maintenance procedures. Noise is the most noticeable indication of bearing failure, but determination of the origin of noise is difficult. Overheating is also a good indicator, but some axles normally operate at temperatures of 200-225 deg F.

Desiderata

Requirements of bearing lubricants include freedom from foaming and corrosion. Lubricants satisfactory for gears are also suitable for bearing lubrication. Dirt or grit in lubricants cause rapid bearing wear. Cylindrical roller bearings employed in gas turbines operate at D-N values of nearly 2,000,000, temperatures up to 600 deg F, and rubbing velocities of 9,000 ft/min. Their lubricants are required to operate satisfactorily at 700 deg F, and efficient oil feeds to critical surfaces are essential. The author quotes an instance of failure of leaded brass cages at high temperatures, with considerable transfer of brass to the steel cage-guiding surface, and cage seizure. Lubrication feed was to the side of the bearing away from the turbine wheel, and examination of failed bearings showed that windage caused oil starvation on the side nearer the turbine wheel, with consequent failure of the brass cage.

Frosting

"Frosting" or "smearing" of bearing surfaces, possibly due to fatigue, is associated with high speed and lack of radial load. It is suggested that such smearing may be eliminated by proper lubrication. Some deficiencies have been alleviated by suitable plating, but it is felt that an alternative approach through improved lubricants and materials is preferable. Visual means of detecting deterioration are satisfactory. However, at high rotational speeds, failure can be very rapid. The rubbing velocity of the roller in the cage pocket should not exceed 9,500 ft/min; if it does, excessive cage wear may result. *M.I.R.A. Abstract No.* 6755.

MACHINE TOOL CONTROL

An Interesting Development by Alfred Herbert Ltd.

IN the past few years there has been a renewed interest in the possibility of controlling machine tools by means of tape or film. The basic principle involved is not new, in fact it was about 1804 that J. M. Jacquard invented a loom that was controlled by punched cards linked together in a chain, but it is only recently that tape or film control has become practical for machine tools. To-day experiments are being carried out with several different systems, and one of the most interesting and promising is the magnetic tape control system developed in the Factored Division of Alfred Herbert Ltd., Coventry.

Fig. 1. Herbert magnetic-tape control equipment

At the outset it may be as well to describe briefly the principles involved in making a sound record. When sound impinges on a microphone, the vibrations produced cause very small electrical currents to flow; these are amplified and fed to an electro-magnetic recording head. If a plastic tape with a metal oxide coating on one side is passed at a constant speed through the varying magnetic field of the recording head, the tape becomes magnetized in proportion to the intensity of the original sounds. On playback, the magnetized tape passes the playback head and induces varying currents that are amplified, fed to a loud speaker, and appear as sounds again.

If some method is devised for converting machine movements to such a form that the magnetic field in the recording head can be varied in a fixed relation to the original movements, the preservation and reproduction of a machine cycle is as easily dealt with as music or speech. Various systems, such as the use of the unusual characteristics of Selsyn motors are being investigated to give electrical signals of positions and speeds. The Selsyn motor can act as a "generator" and when its shaft is rotated by some external power, a signal which is a reference to its angular position is produced. If a servo motor drives a machine slide, which in turn is driving a Selsyn motor, then by means of auxiliary apparatus in which the Selsyn signal is balanced, the servo motor can be made to run only so long as a difference exists between the signal from the Selsyn motor and reference in the auxiliary apparatus. As soon as the difference is eliminated, the servo motor will stop. Accuracy to within 10 minutes' angular rotation can be maintained. With this system the recorder "memorises" the patterns and measurements of movement, but the amount of equipment involved and the complexity of the tape recorder may not be justified by the results obtained. In addition, with this system it is essential to record the movement of each machine slide or individual operation on a separate track of the tape.

Another method of control entails the reduction of the machine functions to coded data by digital computors, whence it is fed into the recorder as a series of pulses. With this system the loss of one pulse out of perhaps hundreds of thousands may destroy, or at least produce inaccuracies in, the work.

In the system, covered by patent application, with which Alfred Herbert Ltd. have been experimenting, the equipment is kept within reasonable proportions and the risk of losing a signal is eliminated. Furthermore, the recorder is simpler than the standard commercial units. It should be pointed out that the recorder was originally developed faithfully to record complex combinations of frequencies, such as those from a large orchestra, and all are intermingled on only one track. Thus machine movements are converted into frequencies that can be heard as musical tones if the output from the recorder is applied to a loudspeaker.

The prototype magnetic-tape control equipment used by Alfred Herbert Ltd. is shown in Fig. 1; the recorder is a standard commercial unit. This has been applied to the Heald Borematic shown in Fig. 2. Most machines of this type have one cycle of operations, and the pattern of the machine functions is normally obtained by conventional electrical methods. Quite often, the cycles necessitate a fairly intricate circuit and a large number of limit switches and relays. Obviously, the more complicated the circuit, the more difficult it is to change or vary the machine cycle. The magnetic-tape control equipment developed by Alfred Herbert Ltd. reduces the work involved in changing cycles to an absolute minimum and any cycle within the capabilities of the slides and elements of the Heald machine

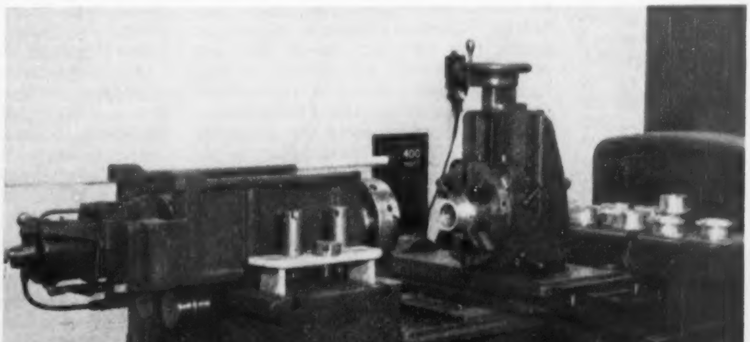


Fig. 2. A Heald Borematic operated by a magnetic-tape control that provides three cycles

can be reproduced without any alteration to the control circuit of the machine. Only the length of tape accommodated on a standard reel limits the length of cycle that can be recorded.

Conversion of the machine was simple. All the limit switches and relays were removed, and only the main hydraulic motor and starter and the solenoid valves were retained. The solenoid valves were connected to slave relays that are actuated by the tuned reeds or tuned frequency circuits. Each of the reed relays or tuned frequency circuits is responsive to only one frequency, and each machine function has an individual controlling relay. The frequencies are generated separately by oscillators that work continuously. When a certain machine function is required, the appropriate push-button is pressed; this allows an electrical feed of the correct frequency to complete the circuit to the reed relay, which operates the slave relay controlling the solenoid valve. For the length of time that the button is pressed, the frequency is impressed on the magnetic tape. At the end of the movement, the button is released and the recording is interrupted. This procedure is followed to record all the machine movements.

If the machining operations allow several movements to take place simultaneously, the total combination of frequencies can easily be recorded. When the cycle is completed manually, the signals for each movement are inherently locked together on the tape. Now, if the recorder and the machine are respectively rewound and returned to their starting positions, the recorder can be played back automatically so that the frequency response circuits, after sorting the signals into their correct channels, will operate the solenoid valves in the original manual sequence and so exactly reproduce the operation cycle.

Should a mistake be made while the manual operation is being recorded, the tape and the machine can be returned to the starting position, the original signals "wiped out," and a re-recording made. If a mistake occurs at the end of a very long cycle, it is only necessary to "wipe out" the signal of the movement on which the mistake was made. In addition, the tape can be cut and spliced and movements eliminated if they are not required.

It is not necessary to have a separate track for each machine movement; any number of different signals can be accommodated on one track of the tape. It should also be appreciated that if failure occurs in the recording play back circuit or in the tape itself, then all the movements of the machine are stopped, whereas, if the system used separate tracks, it would be possible for the movements to continue without reference to breakdown of, say, the playback head on another track. The actual process of recording automatically erases previous information, and there is no risk of accidentally super-imposing a new cycle on an old one.

On the Heald machine, the extreme accuracy of machining that is required is obtained by the use of dead stops, and the Alfred Herbert system does not depend upon the tape and its associated control for extreme accuracy of movement in relation to the exact time the tape is playing. Therefore, the system is used for recording and reproducing patterns of machine behaviour and not for absolute measurement. The dimensional accuracy of the final machine movement is an inherent function of the machine. Before recording is carried out, the hydraulic circuit is pre-set to give certain rates of feed, and the recorded signal ensures that the slide squeezes against the dead stop with the viscosity of the hydraulic oil at its lowest. It is not economical to make the tape an absolute reference of measurement on any machine where the accuracy can be defined by dead stops. Any type of machine now actuated by electro-responsive devices, such as solenoids, magnetic clutches and servo motors, could be operated by this system.

CORRESPONDENCE

FRONT SUSPENSION

SIR,—I should like to reply to Mr. J. E. Jackson's letter, which appeared in your June issue, in which he states that certain errors were made in my article on divided axle front suspension published in the January issue.

His first point is that camber change is not determined by the height of the pivot position, but by the distance between the pivot point and the tread centre. Obviously the shorter this distance, the greater the change in camber angle for a given wheel deflection; however, it was intended in the article to show the effect on camber angle of the different positions of the pivot point in the vertical plane, assuming a given track dimension. The text together with the illustration should have made this clear.

In his second point, regarding the Ford front suspension, Mr. Jackson states that the upper pair of lines locating the swing centre of each wheel should be perpendicular to the king pin and not horizontal as shown, and that this would result in a slightly higher roll centre. This statement is not at all clear to me, since any geometrical arrangement must account for the angle of the lateral links which Mr. Jackson does not mention. In this suspension system, the angle that the lateral links make with the ground has a greater influence on the height of the roll centre than any other single factor.

Regarding the third point, that is, the "sprag effect," it is admitted that this should have been mentioned, but its effects are nothing like so serious as Mr. Jackson suggests. The effects of changes in tyre adhesion and the sprag action combine to prevent a tyre coefficient of unity from being obtained. In any case, the coefficient is not as high as this under normal conditions: the value of unity was assumed in the article only as a basis for comparison of roll angles. When the coefficient is modified by the change in wheel angle resulting from the sprag action, the upward forces are reduced to such an extent that they are not in any way objectionable or detrimental to the handling characteristics of the vehicle. On cars such as the Allard Palm Beach, with a preponderance of the weight on the front axle, the sprag effect is not noticeable under even the hardest cornering conditions.

An understeer effect is not so desirable in vehicles with a preponderance of their weight on the rear axle as in those with the opposite weight distribution characteristics. When the greater proportion of the weight is on the rear axle, it is more difficult to obtain balanced cornering characteristics. In these circumstances the tendency is for the vehicle to begin by understeering into a corner until the centrifugal force overcomes the tyre adhesion at the rear and causes what is commonly known as "breakaway." This is, of course, most undesirable.

In his paragraph referring to the comparison of wheel angles, Mr. Jackson mentions a vehicle with a front suspension having equal and parallel links and a roll centre of 11 in. Such an arrangement, with 3 in rebound, would result in an excessive lateral scrub of about 5 per cent track variation, as compared with 1 per cent for a single link arrangement. The double link layout would also have the disadvantage that the angle of the wheels would always be equal to the roll angle of the body. Also the roll centre becomes progressively lower as the roll progresses, thus reducing roll resistance when it is most needed.

Mr. Jackson considers it unfortunate that most modern cars have a preponderance of weight on the front axle. However, I consider that for a two-seater sports or racing car the weight distribution should be either equal, or with a slight preponderance on the front axle with suspension characteristics to suit.

D. R. HUME.

[We agree with Mr. Hume's comments on the first point. In general, we concur with Mr. Jackson on the second point, but at the same time, Mr. Hume's remarks concerning the influence of the attitude of the lateral links on the height of the roll centre are, of course, correct too. The comments in this letter concerning understeer effects in vehicles having a preponderance of weight on the rear axle are worthy of note. This correspondence must now be regarded as closed.]

PISTON TEMPERATURES

A Method of Calculation for Water-Cooled Petrol Engine Units

Dr. Ing. A. I. Ibrahim Abdelfattah*

TEMPERATURES that will be reached in operation and heat flow control are important considerations in piston design, and much of the improvement in piston efficiency during the last decade can be ascribed to careful study of piston temperatures from which more accurate estimation of expansion effects can be made. Several methods have been adopted for measuring piston temperatures. One is the calorimetric method in which small fusible plugs of known melting points are inserted in the piston surfaces.[1] Another is the recovery hardness method.[2] In this, after a standard running period of 50 hours, the Brinell hardness at selected spots is compared with a known standard to give an indication of the local temperatures reached during the test. A third method is to estimate temperatures by comparing the grain structure of etched sections with known standards. Use has also been made of thermocouples to give direct readings of temperatures, but the high speeds involved make this method difficult to apply.

These notes deal with the estimation of piston temperatures and describe a method by which they may be calculated. Later, an application is given together with a comparison between aluminium alloy and cast iron pistons. At the outset, however, it is as well to consider the factors that govern piston temperatures in automobile petrol engines. There are three types of material in use: cast iron, Y-alloy and a low-expansion aluminium alloy containing about 10 per cent silicon.

Of these three materials, the low-expansion alloy is to be preferred for pistons for two reasons. It is a better material for die-casting and production costs are low, and functionally it is better in that cold clearance can be smaller because of the low rate of thermal expansion. The melting point of this alloy is approximately 537 deg C (1000 deg F), and its thermal conductivity is approximately 120 B.T.U/ft-hr-deg F (180 k cal/m-hr-deg C). For this alloy the maximum piston temperature should certainly be kept below 400 deg C (750 deg F), since the physical strength decreases appreciably at higher temperatures. In fact, the maximum crown temperature should not exceed 375 deg C (706 deg F) for overloads and 350 deg C (660 deg F) for continuous rating.

Various factors affect piston temperatures. For example, they are greater at higher engine loads and mean speeds. The fuel used also has an effect, and the higher the heating value of the fuel the higher will be the piston temperature. In addition, early ignition will lead to increase in piston temperature. Temperatures are also influenced by dimensions: by thickening the crown, the crown temperature can be lowered by as much as 30 deg C (54 deg F) but the skirt temperature will increase by about 15-20 deg C (27-36 deg F). Immediately below the exhaust valve, the piston temperature is often about 10-15 deg C (18-27 deg F) higher than over the rest of the crown.

* Engineering Faculty, Alexandria University.

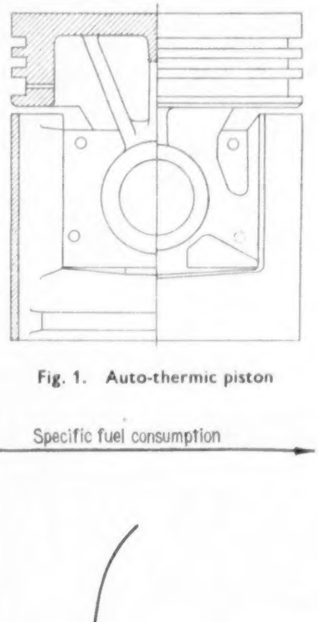


Fig. 1. Auto-thermic piston

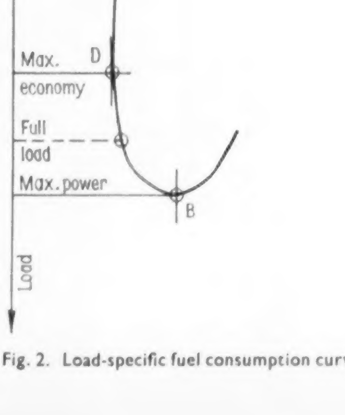


Fig. 2. Load-specific fuel consumption curve

The rate of heat transfer from the piston to the cylinder wall is also important, since the higher the rate the greater the heat dissipation and the less the rate of build-up of piston temperature. Theoretically, the clearance between the piston and the cylinder wall is filled with lubricating oil with a conductivity in the order of 0·07 B.T.U/ft-hr-deg F, and the amount of heat transferred from the pistons to the liner depends to a great extent upon the thickness and viscosity of the oil film, both of which are affected in some degree by the piston temperature. In general, it may be said that the greater the clearance the higher the piston temperature, and at light loads the heat dissipation from the piston will be relatively bad.

There are two points of view regarding heat flow through the piston. One theory is that most of the heat transfer from the piston to the cylinder wall is through the skirt. Three reasons are advanced in support of this theory. First, the aluminium skirt has a thermal conductivity three times that of iron, the ring material; secondly, the area of contact between the skirt and the cylinder wall is far greater than that between the rings and the cylinder wall; and thirdly, the rings do not have metal-to-metal contact with the wall, but are always floating. The other theory is that most of the heat dissipation is through the rings. This point of view is supported by the fact that the pressure between the ring and the cylinder is greater than that between the skirt and the wall. Moreover, it is claimed that active oil films are very effective heat convectors.

In good engine designs, the top of the water jacket is kept higher than the upper piston ring at top dead centre. This causes the flow lines to gather at the ring belt, and thus the theory that about 70 per cent of the heat flow from the crown flows through the rings can be regarded as true. Within limits, increasing the number of rings decreases the piston temperatures, but the effect of each successive ring is much less than that of the ring above. In fact, the effect of the uppermost ring is very much greater than the combined effect of the other rings. The thicker the ring the greater the heat flow and therefore, the lower the piston temperature. However, the temperature at the ring belt should not exceed 212-235 deg C (413-454 deg F) according to the lubricant, in order to maintain the film of oil necessary for lubrication. The ideal to be sought is an even temperature distribution over the whole crown in order to avoid distortion, since distortion can be serious if it reaches the ring belt.

Modifications such as the split skirt and the T slotted skirt have been adopted to improve piston performance.

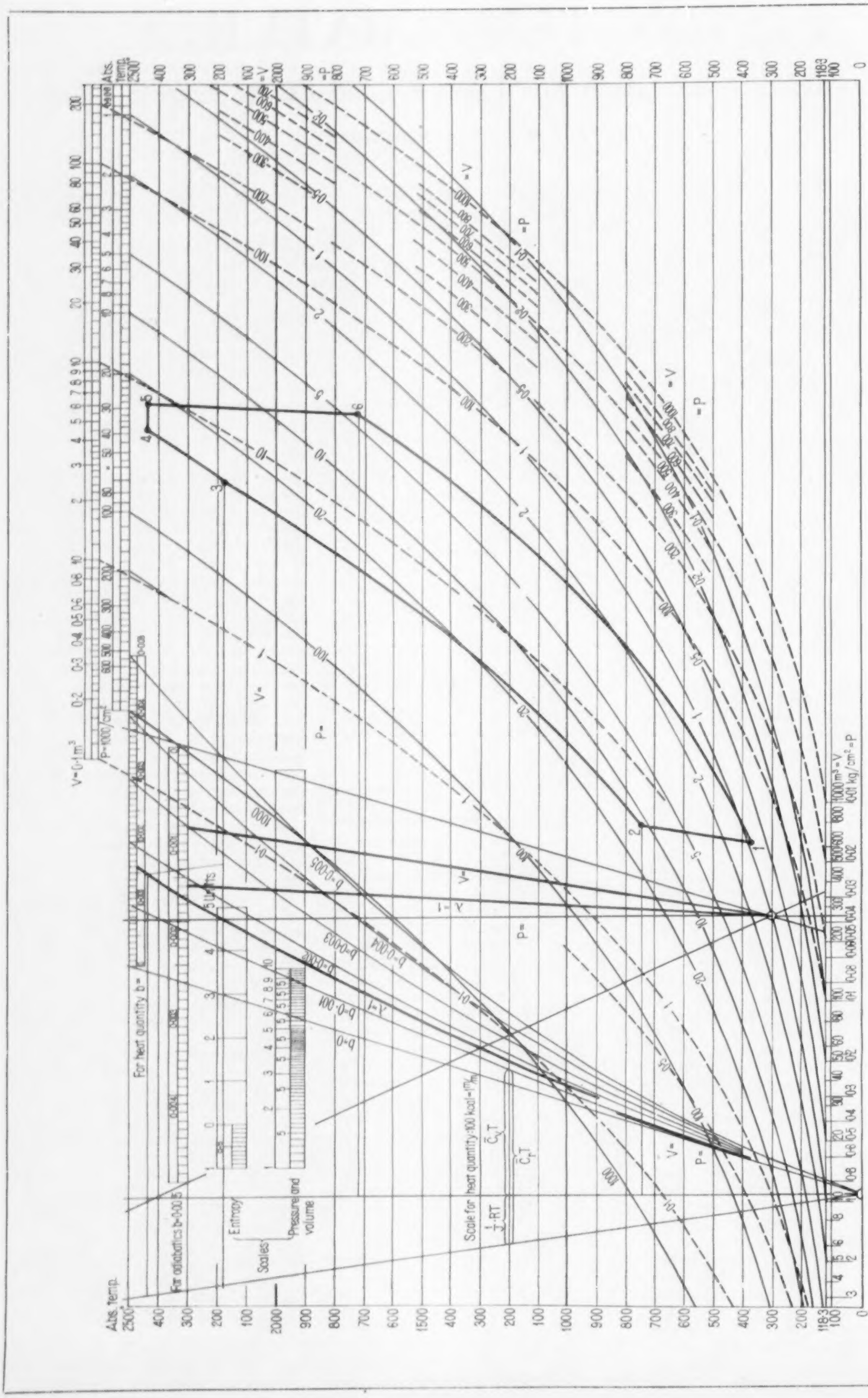


Fig. 3. The Stodola chart

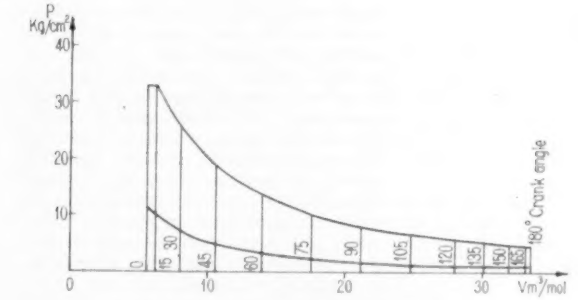


Fig. 4. Calculated indicator diagram

These modifications undoubtedly allow much closer cold clearances to be employed and give silent running both when the engine is cold and at normal operating temperatures. The full-split skirt is favoured because it adapts itself to worn cylinder bores and alignment is not critical. More recently, the autothermic piston has been developed. In this, the aluminium head is mounted on the skirt with an invar steel fitting between to serve as an insulator. The skirt is lead coated to prevent pitting as a result of cold starting. When a piston of this type is exposed to very high temperatures, the heat losses are relatively smaller and thus the thermal efficiency of the engine is increased.

Heat transmission from combustion gases to piston crown

The problem of heat transfer from the combustion gases to the combustion chamber walls in water-cooled petrol engines has been investigated by several experimenters. Nusselt[3] expresses the heat transfer coefficient, h, in the formula

$$h = h_c + h_r$$

where h_c relates to the convection and h_r to the radiation heat transfer coefficients in metric units. He further gives that

$$h_c = 0{\cdot}99\sqrt[3]{p^2.T}(1 + 1{\cdot}24\bar{c})$$

and

$$h_r = 0{\cdot}362\ \left\{\left(\frac{T}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4\right\}\ /(T - T_w)\ \text{(metric units)}$$

Calculations to determine the heat transfer coefficient have been carried out by Janeway.[4] For a water-cooled automobile engine with a 3¼ in × 4¼ in L head, a compression ratio of 5·4 : 1 and a piston area of 8·3 in², he gives values ranging from 0·2 to 0·45 B.T.U/in²-hr-deg F depending upon the charge weight per unit time.

Perhaps the most difficult value to assess is the mean effective gas temperature of the working fluid. Some investigators have tried to determine the effective value by varying the coolant temperature and extrapolating to determine the coolant temperature for zero heat flow.[5] Pinkel[5] for chemically correct mixtures gives an effective value of 1200 deg F (650 deg C) near the cylinder head, while Zipkin and Sanders[5] give 2250 deg F (1240 deg C) near the exhaust valve, the coolant temperature being 180-200 deg F (84-95 deg C).

Opel Olympia 1952 engine

To illustrate the theory on which these notes are based, a detailed exposition will be given of the calculations to determine the temperatures and flow lines for the piston in the Opel Olympia 1952 engine. The specification for this 1·5 litre engine is:—

No. of cylinders	4
Bore	3·1496 in
Stroke	2·9133 in
Piston displacement	90·1 in³
b.h.p.	46 at 3,700 r.p.m.
Maximum torque	68·7 lb-ft at 2000 r.p.m.
Compression ratio	6·15 : 1
Mean piston speed	1800 ft/min

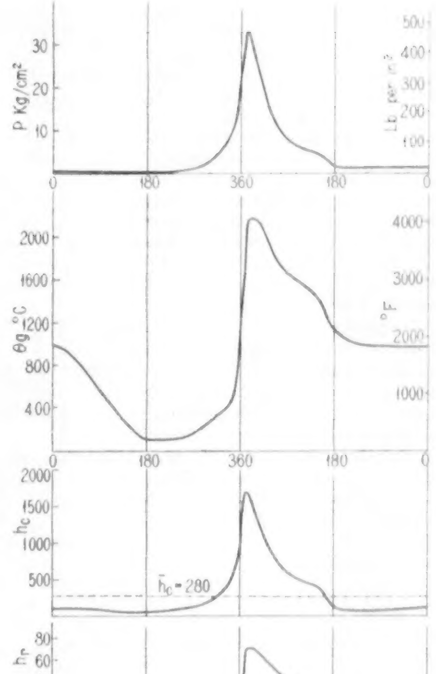


Fig. 5. Pressure-crank angle, temperature-crank angle, h convection-crank angle, h radiation-crank angle, $h\theta_g$-crank angle diagrams

The piston, which is shown in Fig. 1, is of the autothermic type and is made of aluminium alloy. The thermal conductivity of the crown material is about 100 B.T.U/ft-hr-deg F. For the purposes of the calculations it is first necessary to determine the mean effective gas temperature and the heat transfer coefficient from the combustion gases to the piston crown. Both can be determined by using the Nusselt formula in conjunction with the indicator diagram.

An examination of the well-known specific fuel

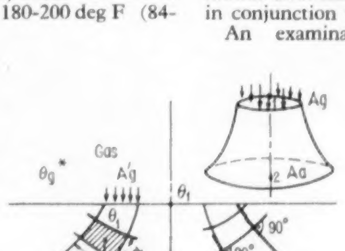


Fig. 6. Tube of heat flow

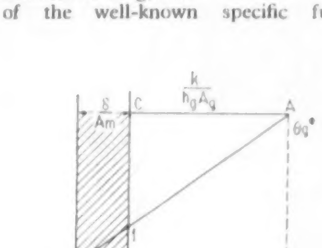


Fig. 7. Graphical determination of crown temperature

consumption curve, see Fig. 2, shows that maximum economy is attained at point A, and maximum power at point B. Now, if the term λ is used to denote the excess air factor, which is defined as the ratio between supplied air and chemically correct air, it may be said that $\lambda = 1{\cdot}1$ at A (lean mixture for maximum economy) and $\lambda = 0{\cdot}9$ at B (rich mixture for maximum power). The position $\lambda = 1{\cdot}0$ (chemically correct mixture) will lie between points A and B on the curve. This condition $\lambda = 1{\cdot}0$ will receive more attention since it resembles the "full-load" condition of the petrol engine.

Commercial benzene can be regarded as containing:

85·3 per cent carbon by weight
14·7 per cent hydrogen by weight.

It has a lower calorific value of about 18,000 B.T.U/lb. With $\lambda = 1$, 1 lb of benzene requires about 15 lb of air for complete combustion, and the products of combustion will contain:

CO_2	0·0713 mol
H_2O	0·0740 mol
N	0·4070 mol
	0·55 mol combustion gases.

Thus the calorific value of the combustion gases will be about 36,000 B.T.U/mol (20,000 k cal/mol). It is assumed that cooling water loss will absorb 27 per cent of this, and of the remaining 14,600 k cal/mol, 70 per cent will be added to the cycle at constant volume, 20 per cent at constant pressure and 10 per cent at constant temperature.

From the known and assumed conditions, the petrol engine cycle may be plotted on the Stodola chart, Fig. 3. On this chart the molecular specific heat is expressed in the form:

$$C = a + bT$$

where 'a' is a constant for all gases, while 'b' is a variable. For pure air 'b' = 0·0008.

During the compression stroke there is a mixture of air, benzene and combustion gases, the resultant 'b' for this mixture may be taken as 0·0009. At the completion of combustion and for "full load" conditions, the value of 'b' is 0·0016. Therefore, on the Stodola chart, lines for 'b' = 0·0009 and 0·0016 can be drawn for adiabatics and heat quantities.

When compression starts the temperature is 100 deg C (212 deg F) and the pressure 0·9 kg/cm² (12·8 lb/in²); the specific volume V_1 and the internal energy U_1 are known and give a point of origin. From this point a line is drawn parallel to the adiabatic line 'b' = 0·0009 and through the known specific volume V_2, which equals V_1/r, a point is fixed on this line. The internal energy established by this point is U_2. The amount of heat added at constant volume plus U_2 establishes a third point. The total heat at the third point, which is derived from the chart, plus the heat added at constant pressure determines the fourth point. The heat added at constant temperature when divided by the absolute temperature T_4 gives the difference in entropy, from which the fifth point is fixed. Finally, the adiabatic through the fifth point and parallel to the adiabatic line 'b' = 0·0016 intersects the constant volume line V_1 at the sixth point to establish the internal energy, U_6.

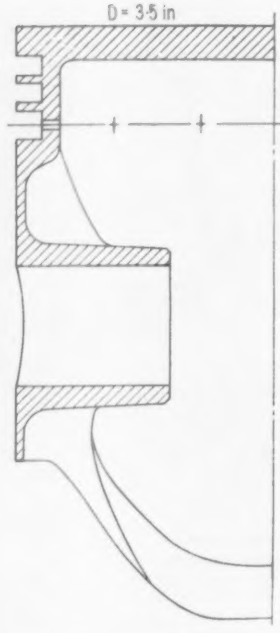


Fig. 9. Chevrolet piston

From the Stodola chart it follows that the heat rejected will be $U_6 - U_1 =$ 8,500 k cal/mol. As the heat added is 14,600 k cal/mol, it follows that the work done equals 6,100 k cal/mol. Now $V_1 - V_2 = 33{\cdot}5 - 5{\cdot}5 = 28$ m³/mol, therefore the indicated mean effective pressure will be:

$$\overline{P}_i = 9{\cdot}3 \text{ kg/cm}^2 \ (132 \text{ lb/in}^2).$$

Assuming a mechanical efficiency of 80 per cent, the brake mean effective pressure will be:

$$\overline{P}_b = 7{\cdot}4 \text{ kg/cm}^2 \ (105 \text{ lb/in}^2).$$

The brake thermal efficiency will amount to 24·5 per cent, the cooling water loss to 27·0 per cent and the rest to 48·5 per cent. Since the engine develops 46 b.h.p. at 3,700 r.p.m., then the b.m.e.p. = 7·42 kg/cm² (105 lb/in²), which justifies the assumptions and calculations. Fig. 4 shows the expected theoretical indicator diagram, while the pressure-crank-angle diagram and the temperatures taken from the Stodola chart are given in Fig. 5. The maximum pressure in the combustion chamber reaches about 33 kg/cm² (467 lb/in²) and the maximum temperature is about 2180 deg C (3950 deg F).

Application of the Nusselt formula

From the Nusselt formula quoted earlier, the coefficients of heat transfer through radiation (h_r) and convection (h_c) were calculated for each position; then the mean values were determined by planimetering. The calculations give:

$h_c = 0{\cdot}397$ B.T.U/in²-hr-deg F (280 k cal/m²-hr-deg C)
$h_r = 0{\cdot}022$ B.T.U/in²-hr-deg F (15·6 k cal/m²-hr-deg C)

Therefore the total coefficient of heat transfer will be:

$h = 0{\cdot}419$ B.T.U/in²-hr-deg F (295·6 k cal/m²-hr-deg C).

By multiplying the heat transfer coefficient, h, by the gas temperature θ_g, and drawing a curve ($h.\theta_g$) against the crank-angle ϕ, the mean value of $h.\theta_g$ is obtained. From this the mean effective gas temperature (see 3), sometimes called

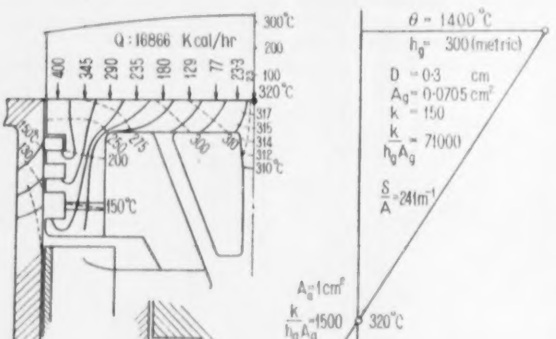


Fig. 8. Isothermals for auto-thermic piston

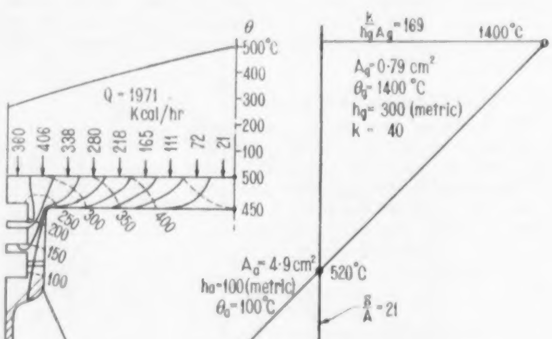


Fig. 10. Isothermals for Chevrolet piston

the resultant gas temperature, can be obtained. It is designated θ_g and is:

$$\theta_g = 2640 \text{ deg F } (1450 \text{ deg C}).$$

Comparison shows that the value obtained from the Nusselt formula for the heat transfer coefficient, h, from the combustion gases to the piston crown is in substantial agreement with the value quoted by Taylor.[5] On the other hand, the mean effective gas pressure derived from the Nusselt formula is higher than stated by Taylor. This is probably due to the fact that Taylor's figures are based on experimental data and the test mean piston speeds are lower than in the calculated cases, where they reach 1,800 ft/min. From calculations based on the Eichelberg equation[6]

$$h = 2{\cdot}1.\sqrt[3]{c}\sqrt{p.T} \text{ (metric)},$$

the following values were arrived at:

$h = 0{\cdot}387$ B.T.U/in^2-hr-deg F (272 k cal/m^2-hr-deg C)

and

$\theta_g = 2427$ deg F (1330 deg C).

For calculating the piston temperatures, the author uses the following values:

$h = 0{\cdot}427$ B.T.U/in^2-hr-deg F (300 k cal/m^2-hr-deg C)

$\theta_g = 2532$ deg F (1400 deg C).

It is also necessary to consider conditions in the crankcase. During running, the crankcase will contain hot air together with some lubricating oil vapours. Therefore, one side of the piston is exposed to combustion gases and the other to hot air and oil vapour, whose temperature reaches about 80 deg C. The ribs connecting the piston body to the crown will help heat transfer from the gases in the crankcase. For this, the coefficient of heat transfer is taken as:

$h = 0{\cdot}1425$ B.T.U/in^2-hr-deg F (100 k cal/m^2-hr-deg C).

The piston can be regarded as a disc attached to a cylindrical barrel. On one side, there are combustion gases of mean effective temperature θ_g and rate of heat transfer h_g. On the other side there are hot air and vapours, to which heat from the piston crown is transferred at a rate h_a. The quantity of heat Q passing through the surface area A_g will tend to go through the tube of flow shown in Fig. 6, and then out of the area A_a to the hot air, whose temperature is θ_a. If the temperatures at the piston centre, area A_g, be θ_1, and on the air side be θ_2, then

$$Q = h_g A_g (\theta_g - \theta_1)$$
$$= h_a A_a (\theta_2 - \theta_a)$$

Taking the thermal conductivity of the metal crown as k, crown thickness δ, and A_m the mean cross sectional area,

$$A_m = \tfrac{1}{2}(A_g + A_a)$$
$$Q = kA_m(\theta_1 - \theta_2)/\delta$$

Eliminating θ_1 and θ_2 between the three equations:

$$Q\,(1/h_g A_g + \delta/k A_m + 1/h_a A_a) = \theta_g - \theta_a$$

putting

$$\frac{1}{h_g A_g} + \frac{\delta}{k A_m} + \frac{1}{h_a A_a} = \frac{1}{K}$$

Then

$$Q = K(\theta_g - \theta_a)$$

where K = rate of heat transmission from combustion gases to the hot air.

The heat quantity can be written as follows:

$$Q = k \tan \epsilon$$

$$\text{where } \tan \epsilon = \frac{\theta_g - \theta_a}{\dfrac{k}{h_g A_g} + \dfrac{\delta}{A_m} + \dfrac{k}{h_a A_a}}$$

A graphical representation of the problem is shown in Fig. 7. AC is taken $= k/h_g A_g$ and represents θ_g, BD $= k/h_a A_a$ and represents θ_a. Joining AB, it cuts the apparent crown thickness at point I representing the surface temperatures θ_1 on the gas side. This merely gives an idea of the piston crown temperature at the mid-centre, θ_1.

Now, the amount of heat flowing through the tube of flow shown in Fig. 6 is:

$$Q = h.A_g.(\theta_g - \theta_1).$$

This amount will be assumed flowing through the area:

$$2\pi r.\Delta b, \text{ where}$$
$$Q = k 2\pi r \Delta b.\Delta\theta/\Delta a$$

For any given Q and temperature drop, the distance 'a' can be calculated.

A graphical estimation of the temperature θ_1 at the piston crown centre is given in Fig. 8. It should be mentioned that the solution of the problem starts with an assumption of a central flow tube, which suits the heat flow conditions. The surface temperature θ_1 reaches 320 deg C (607 deg F).

Then, if the piston crown thickness is divided into very small heights, the equation

$$Q = \pi d^2.k.\frac{\Delta\theta}{\Delta y}$$

is applied to estimate the temperatures at the central section of the crown. The calculation gives a difference of 10 deg C (18 deg F) at the crown centre, which is very small. Behind the first ring, the temperature reaches 200 deg C (392 deg F).

The heat flow lines at the rings were estimated after making the following assumptions as regards heat transfer from:

Ring to cylinder	h = 35,000 k cal/m^2-hr-deg C
	= 50 B.T.U/in^2-hr-deg F
Piston and cylinder	= 500 k cal/m^2-hr-deg C
	= 0·715 B.T.U/in^2-hr-deg F
Piston and upper side of ring	= 500 k cal/m^2-hr-deg C
	= 0·715 B.T.U/in^2-hr-deg F
Piston and lower side of ring	= 15,000 k cal/m^2-hr-deg C
	= 21·5 B.T.U/in^2-hr-deg F.

The temperature at the lower side of the upper piston ring rises to 170 deg C (338 deg F), at the contact surface with the cylinder 165 deg C (329 deg F), while the temperature of the cylinder itself rises to 150 deg C (302 deg F).

The construction of the flow lines had to be corrected several times to suit the running conditions of the engine. Then the contours, Fig. 8, indicated that about 68 per cent of the heat was transmitted from the piston crown to the cylinder liner through the piston rings, 14 per cent through the piston skirt to the cylinder, and the remaining 18 per cent to the hot air and vapours in the closed crankcase. Because of its good conductivity, the aluminium alloy piston crown causes a drop in temperature from the crown centre to its end of about 320 − 250 = 70 deg C (126 deg F).

The Chevrolet 1951 automobile engine piston is shown in Fig. 9. It is made of cast iron. In what follows it is assumed that the effective gas temperature and rate of heat transfer are the same as in the case of the Opel piston; the same heat transfer from crown to gases in the crankcase is also assumed, but the temperature of those gases will be taken as 100 deg C (212 deg F). If the method described earlier is applied, the central crown temperature on the gas side is 520 deg C (967 deg F), and the thermal conductivity of cast iron is 40 k cal/m^2-hr-deg C (27 B.T.U/ft^2-hr-deg F).

Proceeding as described before, the flow lines and piston isothermals can be completed, as shown in Fig. 10, which shows that the temperature drop at the axis of the piston crown amounts to 50 deg C (90 deg F) while the temperature difference between crown thickness and outer diameter amounts to about 270 deg C (485 deg F).

A comparison between the two pistons may be made: The coefficient of expansion of cast iron is $5{\cdot}9 \times 10^{-6}$/deg F and of aluminium alloy 12×10^{-6}/deg F. Comparing the two piston clearances, the diameters being nearly the same, then the ratio: Clearance for cast iron piston, S_{CI}, to aluminium alloy one, S_{AL}:

$$\frac{S_{CI}}{S_{AL}} = \frac{\alpha_{CI} \times \Delta\theta_{CI}}{\alpha_{AL} \times \Delta\theta_{AL}}$$

where α, $\Delta\theta$ are the coefficient of expansion and the temperature difference across the diameter at the piston crown surface in contact with the hot combustion gases. Therefore, the clearance for the cast iron piston will be double that of the aluminium alloy piston.

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A SPECIALIST VEHICLE

A Caravan Incorporating Refrigerated Air Conditioning and Other Amenities Required in the Tropics

BECAUSE of its specialist nature, caravan design is not a subject that is normally of much interest to automobile coachbuilders. However, a model recently introduced by Pilgrim Mobile Units Ltd., of Ringwood, Hants., is an exception, because many of its features might be incorporated in other types of coach built bodies. It has been designed specifically for service in hot countries. For such service the main requirements are: adequate insulation against heat from the sun, air conditioning, proofing against the entry of mosquitos and other flies, the use of materials not subject to attack by white ants and other insects, and design to discourage the growth of fungi on the vehicle components. Another important feature of these designs is the provision of washing, or better still, bathing facilities, and of adequate storage space for water. The vehicle must also be sturdy enough to travel long distances at reasonable speeds over rough tracks.

The overall dimensions of the Pilgrim body are: length, including the tow-bar, 18 ft 9 in, width 7 ft 6 in, height 8 ft 9 in. Its weight is 1 ton 8 cwt. Inside the body the dimensions are: 16 ft 8 in long by 6 ft 10 in wide and 6 ft 4 in head room. The body is mounted on a Carlight frame of angle section. A single, straight axle of 1¼ in square section is employed and the wheel hubs are mounted on taper roller bearings. Michelin heavy duty 7.00 × 16.00 tyres are fitted and Girling brakes with 10 in diameter drums are employed. The brakes are operated by an over-run mechanism and by a hand lever for parking. A universal ball type coupling is employed, and brace operated, fully retractable corner props are fitted.

All joints of the main framework of the body are halved and screwed. The members of the body are of African, insect-resisting hard woods and are chemically treated. The exterior panel is of tempered Masonite, and standard Masonite is employed for the interior panelling.

Fibreglass insulation is packed between the double panelling of the roof, and Isoflex is used in the side and end walls. The floor is of 1 in thick tongued and grooved deal and is pressure impregnated with Celcure. Further protection from the heat of the sun is provided for by a large flysheet, 24 ft × 20 ft, which can be erected on poles in a similar manner to the flysheet of a large tent or a marquee.

The interior furnishings are of considerable interest. They have been arranged so that the vehicle can be adapted for a wide range of uses in remote and isolated sites in undeveloped territories. Oak framing and panelling are used throughout. Two single beds are provided; they can be converted into settees. They have interior sprung mattresses, 6 ft 3 in by 2 ft 3 in, and upholstered backrests. These are arranged on each side at the front of the vehicle. Between them, against the end wall, is a chest of drawers. A folding trestle table can be erected between the settees and the seating space is adequate for four to six people.

In the centre portion of the caravan there are two wardrobes, one against each wall, and a knee-hole desk. This desk has a folding top so that the working area can, if necessary, be doubled to 50 in by 30 in. This is a useful feature for surveyors and civil engineers who require desk space for plans and drawings. There are two large drawers in the desk, and two nesting chairs are provided. Also in this part of the caravan are two lockers and a wall bookcase.

The windows are metal framed, and have external louvres at the top and rubber dust excluders. Each window is fitted with a venetian blind and a gauze flyscreen. The floor covering is of inlaid linoleum and the interior paintwork is in broken white enamel throughout. Roof vents are fitted and are designed in such a manner as to provide through ventilation when the caravan is being towed. The forward facing vent is so situated that it is clear of the dust cloud that rises from the wheels of the towing vehicle. There are two doors and these, as well as the roof vents, are fitted with flyscreens.

A detail point of interest in connection with the flyscreens, which are fitted immediately inside the doors, is that they can be opened both inwards and outwards. The reason for this arrangement is as follows. When the vehicle is stationary during the day, the doors normally are open and the flyscreens closed; so when anyone wants to enter or leave the vehicle, they open the flyscreen outwards and thus any mosquitos or other insects on it are driven out, instead of carried inside. On the other hand, when the outer doors are latched and the occupants inside, for instance at night, access to the latch can only be gained by opening the screen inwards.

An Electrolux, kerosene refrigerator, type LK 300, of 3 ft³ capacity, is installed next to one of the wardrobes. The principle of operation of this type of unit is similar to a gas one, except that the heating element burns kerosene. The air carrying the heat extracted from the refrigerated compartment is ducted to the outside of the vehicle. When the caravan is in motion, the duct can be closed by a simple, hinged flap so that dust cannot enter.

At the front, an Arcon ½ h.p. air-conditioning unit is fitted in the window aperture. This unit has a capacity of 6,100 B.Th.U/hr when cooling or de-humidifying the air in the caravan. It can be controlled to cool incoming fresh air, or recirculated air, or a mixture of both. The air intake incorporates a filter unit for removing dust and flies.

At the rear, separated from the main part of the caravan by a sliding door that can be bolted from the saloon side, is the kitchen. The equipment in it includes a food cupboard, stowage for cutlery and crockery, and a two-burner, pressure type, kerosene cooker, as well as a sink unit with a tap-type plunger pump and 12 gallon water storage tank. A filler cap for this tank projects outside the vehicle.

Also at the rear end of the caravan, but beside the kitchen, is the toilet compartment. It is fitted with a wash basin with a tap-type pump that draws water from another storage tank, which is of 15 gallons capacity. This tank is in a locker under one of the settees. Access to the filler tube is also gained from outside the vehicle. In the toilet compartment, a shower rose is fitted and is served by a Rocker pump. The water from the shower runs away through a drain in the floor. This compartment also contains an Elsan chemical closet.

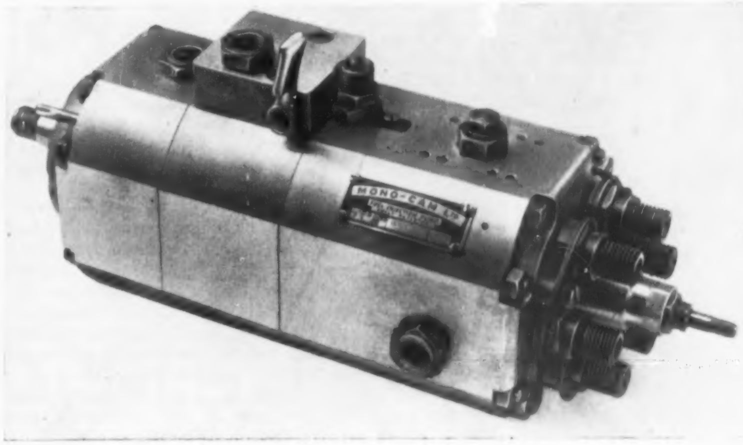
Five 12-volt roof lamps are served by suitably connected 6-volt, 112 amp-hr Tungsten batteries slung in a carrier under the floor, near the rear axle. They are served by a charging unit in the 250-volt power supply circuit. Three wall sockets are connected to the power supply; one supplies current to the air conditioning unit and the other two are for general use for electric razors, kettles, etc. The supply can be taken from 220-250-volt mains or the generator.

A self-contained, portable petrol-driven generator is supplied with each caravan and stowed aboard for travelling. When on site, this generator is unloaded and removed 50 yards from the vehicle where the noise it makes is less likely to be troublesome. It is connected to the caravan by a length of cable. The generator has an output of 1½ kW and is powered by a Norman, flat, twin-cylinder petrol engine. It supplies adequate power for the air conditioner and for charging the batteries, so that electric light is available at night-time without running the generator. An instrument is mounted in a prominent position to indicate the charge state of the batteries.

MONO-CAM INJECTION PUMP

A Newly Developed, Compact, Light-Weight Unit that Gives Good Combustion Characteristics

AFTER seven years of development work, a new injection pump has been introduced by Mono-Cam Ltd., of 28, Queen Anne's Gate, S.W.1. Two types of pump are available, each for either four- or six-cylinder engines. Type A incorporates elements of 5-7 mm diameter: it is about 8 in long and the diameter of its casing is 92 mm ($3\frac{3}{4}$ in). The weight of this unit is 13 lb. Type B incorporates elements of 7-10 mm diameter and its overall dimensions are $10\frac{1}{2}$ in long by 112 mm ($4\frac{1}{4}$ in) diameter. This pump weighs $17\frac{1}{2}$ lb, and an injection unit of more conventional form designed for the same duty weighs about 46 lb.



The Mono-Cam injection pump is a compact and light unit

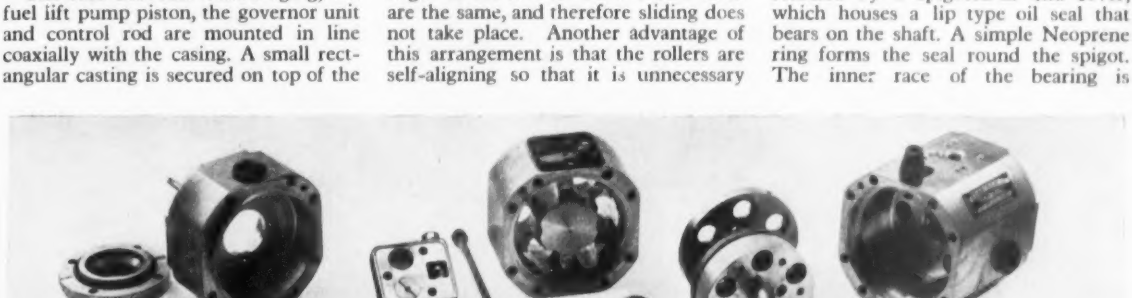
In both types, the housing is in three parts spigoted together in line: the centre one carries the fuel lift pump and roller followers, another houses the governor and injection elements, and in the third is the face cam assembly. When bolted together, these three parts form two chambers, one filled with oil at spill pressure and the other with oil at lift pump delivery pressure. The barrel and plunger elements are in the higher pressure section. They are arranged round the control rod and governor unit and their axes are parallel to the longitudinal axis of the casing. The low pressure section of the unit houses the face cam disc and roller followers that actuate the plungers, together with a fuel lift pump. Sealing of the casing is effected by a Neoprene ring round each spigot.

The face cam and shaft forging, the fuel lift pump piston, the governor unit and control rod are mounted in line coaxially with the casing. A small rectangular casting is secured on top of the centre portion of the casing. It houses the disc type inlet valve and the ball type outlet valve for the lift pump.

Cam and followers

The case-hardened, 5 per cent chromium nickel steel face cam disc is integral with its shaft and the outer periphery of its face is bevelled. This bevel forms the track on which taper roller followers run. The bevel arrangement has been adopted so that the relative velocities of the outer and inner edges of the rollers and the cam track are the same, and therefore sliding does not take place. Another advantage of this arrangement is that the rollers are self-aligning so that it is unnecessary to provide a positive means of locating their carriers against rotation. However, to prevent incorrect assembly, one arm of the forked carrier is extended to overlap the periphery of the cam disc.

A ball thrust race interposed between the cam disc and the end of the casing takes the axial load. Another ball bearing round the shaft takes the journal loading. This bearing is inserted into a housing counterbored from the outer end of the casing. Its outer race is retained by a spigoted-in end cover, which houses a lip type oil seal that bears on the shaft. A simple Neoprene ring forms the seal round the spigot. The inner race of the bearing is



Some of the main components of the Mono-Cam pump, including the rigid three-piece casing and the combined pneumatic-and-hydraulic governor unit

retained by a ring nut screwed on the shaft and locked by a tab washer. This nut applies pre-load to the two ball bearings.

Undoubtedly, the rigidity of this arrangement contributes much to the relatively quiet and vibrationless running of the unit. Since a single cam is employed and it actuates each of the plungers in turn, an identical reciprocating motion is imparted to each plunger. The profile of the cam is designed to give the best injection characteristics for normal requirements, but if necessary special profiles can be employed to suit different combustion chamber designs.

The roller followers are carried on needle roller bearings. A hardened steel washer is interposed between the outer face of each roller and the adjacent arm of the forked carrier, or tappet. This washer takes the thrust due to the inclination of the cam track. The induction hardened, roller carriers are of cylindrical form and operate in bores spaced round the longitudinal axis of the light alloy centre section of the casing. These tappets, as well as the ball bearings round the cam unit, are lubricated by the diesel oil in the low pressure chamber in which the assemblies are housed.

Lift pump

Oil is delivered into the low pressure chamber by the fuel lift pump. In the A type unit this pump is actuated directly by an extension of one of the pins of the roller tappets. The larger pump has a separate roller type follower running on the cam track. With this arrangement the outward stroke of the lift pump piston in its steel cylinder is effected by a short tie rod. One end of the tie rod is screwed into a boss under the piston crown. The other end is of T-shape and is passed through a slot in the centre of an actuating lever and then rotated through 90 deg to seat in a groove machined across the lever. This lever is pivoted near one end that has an extension which can be reciprocated by the hand priming device. Its other end is actuated by the tappet or follower.

The return stroke of the piston is effected by a compression spring. This spring also keeps the T-end of the tie rod seated in the groove in the actuating lever. It is located at one end round the boss into which the tie rod is screwed, and its other end bears against a dished plate covering the outer end of the cylinder. There is, of course, a clearance hole in the centre of the plate for the tie rod to pass through.

During the suction stroke, fuel is drawn in through a disc type non-return valve into a space under the block on top of the casing and thence through a drilled passage to the pump. When the piston returns under the action of the spring, the fuel is forced back into the space under the block and thence through a ball type non-return valve, into the end of the low pressure part of the casing where the fuel lift pump actuating lever is housed. From there it passes into the tappet and cam housing and lubricates and cools the rotating and reciprocating parts. The outlet union is on top of this housing, and the fuel is carried through an external pipe to a filter and then, when combined hydraulic and pneumatic governing is employed, to a union on the side of the injection pump casing, near the delivery end. When hydraulic governing alone is used, a blow-off valve is usually fitted in the filter head to return the excess fuel to the tank. From the union on the side of the pump casing, the fuel goes into an annular chamber and thence through the induction ports into the plunger barrels.

Plunger, barrel and delivery valve assemblies

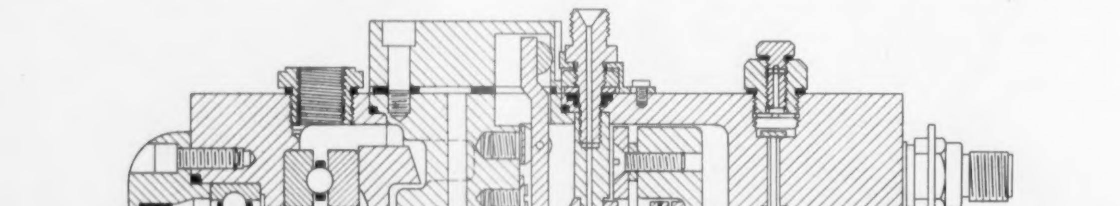
The flanged plunger barrels are assembled into their shouldered housings from the outer end of the high pressure chamber casing. They are retained by the delivery valve assemblies, which are screwed into the housings and bear against the flanged ends of the barrels. The delivery valves are of the pressure release type, and have flat seats to give positive action.

Each valve assembly comprises a valve seating in a thimble shaped housing. A sealing washer is interposed between the thimble end and a flange round the seating. The valve stems are fluted in the usual manner for location in the bore of the seating. Their outer ends are spigoted into the compression spring, and the other end of this spring is located on a shouldered collar. The end of the smaller diameter portion of this collar acts as a stop to limit the valve lift. Accurate control of the lift is important since it influences the torque characteristics of the engine.

Plungers with an unconventional spill device are employed. Their design is such that the suction stroke begins when the flat end of the plunger uncovers the induction ports and, on the delivery stroke, injection begins when the ports are closed again as the plunger moves forward. The unconventional feature is that the spill flows through an axial hole in the plunger and thence radially to an annular groove round its periphery. Spill is rapid, since it takes place when this annular groove moves suddenly out of a sleeve round the plunger. This feature is noteworthy because it results in an appreciable reduction in injector dribble and consequent carbonization.

Spill timing is regulated by moving the sleeves longitudinally. They are housed in holes in a light alloy circular plate. The sleeves are flanged and retained between the plate and the mushroom end of the control rod assembly. Distance pieces between the plate and the mushroom end allow the sleeves a restricted amount of radial float so that they can be freely aligned with the plungers. The thickness of the distance pieces is such that there is no axial float of the sleeves, otherwise the accuracy of the spill timing would be affected.

A Neoprene seal is fitted round the



The longitudinal section of the pump with the combined hydraulic-and-pneumatic governor arrangement

end of the plunger, which projects through the sleeve. It is retained by the plunger return spring and bears against a circular plate that separates the high and low pressure compartments. The coil springs which seat on these sealing rubbers are retained by slotted collars fitted in grooves round the ends of the plungers.

Pneumatic and hydraulic governor

Either hydraulic or hydraulic-and-pneumatic governing can be employed. For vehicle applications, where close governing of the middle range of operating speeds is not essential, the combined hydraulic and pneumatic governing arrangement is employed. The lower end of the range is accurately controlled by manifold vacuum and the upper end by spill pressure.

In principle, the action of both the governors is to regulate the axial position of the mushroom-headed control rod, on which are carried the spill sleeves and their housing plate. The stem end of this rod is carried in a steel bush inserted in the end of the casing. A stud is mounted axially in the end of the stem. Round the stud are the governor and damper springs, and on its end are screwed a lock nut and a retainer collar for the governor and damper springs. The spring assembly is surrounded by a cylindrical housing screwed into the end of the pump casing. The governor spring bears on the end of the housing and loads the rod in such a way as to tend to give maximum delivery. The damper spring bears on a shoulder in the housing and loads the rod in a similar manner but, since it is short, it is only in action at the minimum delivery position. A thimble is screwed over the outer end of the housing and carries the maximum fuel delivery adjustment screw and the plunger for stopping the engine. Additional damping is provided by the fuel oil in the chamber in which is housed the mushroom end of the control rod.

A union for the pneumatic pipe from the manifold is screwed radially through the casing and into the disc that separates the high and low pressure compartments. It communicates with a hole drilled radially in the disc and through a steel bush in its centre. One end of the bush is sealed by a plate secured by screws to the low pressure face of the disc; the other end is closed by a diaphragm. A projection of the control rod forms a guide and slides in the bush. From the end of the guide, an axial hole is drilled and communicates with radial holes. These passages transfer the manifold pressure to one side of a diaphragm, the outer periphery of which is retained between a clamping ring and the rim of a cupped sleeve assembled over the boss in which the bush is housed. The inner periphery of the diaphragm is secured by a clamping ring to a thimble fitting round a shoulder on the mushroom head of the control rod. Thus, the other side of the diaphragm is subject to spill pressure. Neoprene sealing rings are fitted to the plate that closes the end of the bush, the cupped sleeve round the bush housing and to the thimble fitting.

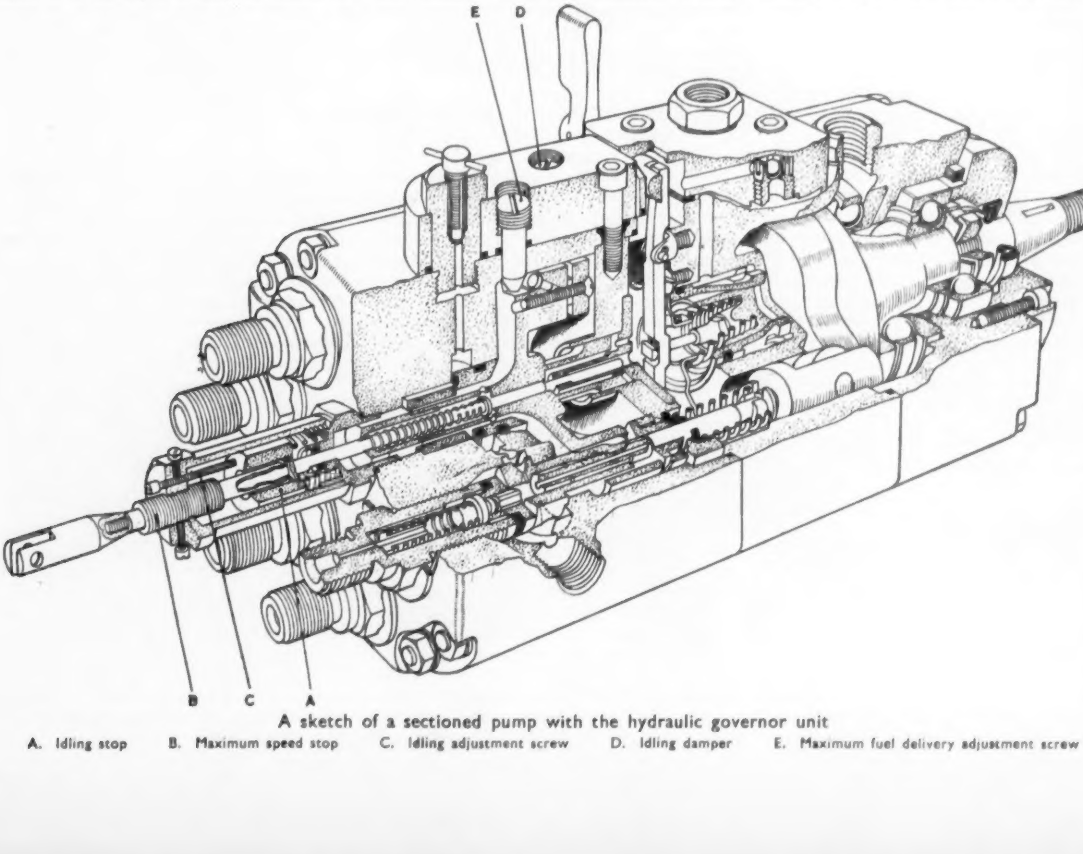
The spilled fuel passes from the high pressure chamber into a steel bush pressed into the end wall of the casing and through a radially drilled hole into the annular delivery chamber round this bush. An aluminium restrictor pin is assembled into the side of the casing and projects into this hole. To adjust the basic setting of the governor, pins of different sizes can be fitted. The differential rates of expansion of the pin and the bush help to counteract the effects of viscosity changes due to temperature variations.

Hydraulic governor

A different arrangement is employed when hydraulic governing only is used. The projection at the centre of the mushroom head of the control rod forms a piston working in a bore in the centre of the plate separating the high and low pressure chambers. The end of the bore is closed by a circular plate as before, but this plate is drilled so that the low pressure is communicated to the space between it and the head of the piston, and a boss at its centre is extended into an axial hole in the piston to form a guide for the control spindle. Spill pressure acts on the stem side of the head of the piston.

From the sketch of the hydraulically governed unit, it can be seen that the spill is returned to the low pressure chamber. This return is through radial holes into the hollow stem of the mushroom-headed control rod, and thence through an axial orifice to the low pressure side of the piston, from whence it goes through the holes in the circular plate that closes the end of the cylinder.

A tapered restrictor pin is carried in the axial orifice in the control rod. This pin is the centre part of a spindle, one end of which, as has already been stated, is supported in the boss in the middle of the plate that closes the end



A sketch of a sectioned pump with the hydraulic governor unit

A. Idling stop B. Maximum speed stop C. Idling adjustment screw D. Idling damper E. Maximum fuel delivery adjustment screw

of the cylinder. The other end of the spindle projects outside the casing of the pump unit. On it is screwed a flanged sleeve of small diameter, and a fork-joint is screwed against the end of the sleeve to lock it in position. Immediately outboard of the restrictor taper, a collar is fitted on the spindle. A compression spring is interposed between this collar and a pierced thimble fitting screwed into the stem of the mushroom-headed rod. The compression in this spring tends to push the mushroom-headed rod, together with the spill sleeves and carrier plate assembly, into the maximum delivery position. Axial movement of the spindle alters the tension in the spring and changes the position of the taper restrictor pin in its orifice.

The pierced end of the thimble, through which the rod passes, is approximately level with the end face of the casing of the pump unit. Into this casing is screwed a cylindrical housing for the idling and maximum speed stops. These stops are positioned one on each side of the flange round the small diameter sleeve screwed on to the end of the spindle, the motion of which they thus restrict. One, the maximum speed stop, is another sleeve passed over the first and screwed into the end of the cylindrical housing of the stop assembly. The other is the minimum speed stop and is a collar, which, to adjust the setting, can be slid longitudinally in the housing. It is moved against the action of a strong compression spring by an adjuster screw in the outer end of the housing. To stop the engine, the rod is pushed inwards against the action of this spring.

Damping is regulated by a screw, marked D in the illustration, in the top of the unit. The inner end of this screw is tapered and projects into the spill chamber. It acts as an adjustable abutment for a spring-loaded plunger housed in the spill sleeve carrier plate assembly. Maximum fuel delivery, governed by the smoke limit, is adjusted by another screw, E. This screw has a conical end which, in the maximum delivery position, abuts against a hardened steel pad in the mushroom head of the control rod.

Advantages

Some of the advantages claimed for this unit have already been mentioned. For example, the large diameter ball thrust race behind the cam track ensures complete rigidity. Another feature that has already been mentioned is that the roller followers are self-aligning on their bevel track.

The type of cam and shaft arrangement employed is relatively free from the torsional and bending vibrations and deflections that may be encountered with the conventional, in-line type of fuel pump. Little wear should be experienced on the moving parts, because all are cooled and lubricated with fuel oil. Since the unit is completely enclosed, there is no danger of foreign matter entering and causing damage.

It is claimed that because of the relatively large amount of fuel spilled as the pump delivery is reduced, the hydraulic governing system gives a sharp over-speed cut-in. There should be little trouble with wear of the control mechanism, which would appear to be better from that point of view than the conventional rack arrangement. Moreover, by the elimination of the helix on the plunger, and consequently of the side thrust, the vulnerability of the unit to wear and the effect of wear on the pump balance, have been reduced, although the ends of the plunger and barrel are just as subject to wear by trapped abrasive particles as are those of more conventional types of plunger and barrel assembly.

Diagrams of the performance of the pump indicate that pilot injection takes place at about 400 r.p.m., but is not maintained over all the range. The general characteristic over the whole load and speed range of operation is of a slow initial rate, followed by a steady delivery of the main charge and ending with a sharp cut-off without after-injection. This gives smooth and quiet combustion characteristics and should make the unit particularly attractive for private car application.

The compactness and light weight of the unit also make it suitable for small installations. Generally, the relative prices of similar engineering products vary approximately in proportion to their weights. Therefore, it seems likely that if produced in large numbers, this pump will be less expensive than units at present used for comparable duties. It should be mentioned that Mono-Cam Ltd. is an associate of Southern Areas Electric Corporation Limited.

ISO-PENTANE

A Development at Stanlow Oil Refinery

THE first plant of its kind to be built in the United Kingdom is now under construction at the Stanlow, Cheshire, oil refinery of The Shell Petroleum Company Limited. It is to be used for the production of iso-pentane, an important constituent of aviation gasoline. One of its columns, believed to be the tallest of its kind in the world, will be 200 ft. high. The main contractors are Head-Wrightson Limited, and the unit is scheduled for completion early in 1955.

Aviation gasoline, which must function efficiently in widely varying climatic conditions, is a blend of highly specialized components of different characteristics, and the iso-pentane from the new plant will replace similar material that hitherto has had to be imported for blending up into finished aviation gasoline in this country. In this respect the new unit will be complementary to the "platformer" at Stanlow, which came into operation last year and by means of which certain gasoline fractions are converted into other valuable aviation blending components. The importance of iso-pentane as an aviation component lies, among other things, in its high octane rating and in its favourable vapour pressure which ensures that the fuel vaporizes efficiently in extremes of temperature.

The new plant is expected to cost nearly £500,000. It will substantially contribute to the versatility of the refining system at Stanlow, already one of the most comprehensive refining complexes in the world. Stanlow, with an output of nearly 5,000,000 tons a year of various oil products, is the largest of the four Shell refineries in the United Kingdom, whose combined annual capacity totals 11,000,000 tons.

In the production of iso-pentane, debutanized light platformate will be led through a steam pre-heater to the depentanizer—an 86 ft. column containing conventional bubble-cap trays—where the pentanes will be removed as a top product. The bottom stream, depentanized light platformate, will be run down to aviation component storage. The de-iso-pentanizer, a 200 ft column containing 140 turbogrid trays, will receive two streams via steam heaters, namely the pentane fraction and debutanized straight-run gasoline. It will produce iso-pentane as the overhead stream. The bottom product, consisting of the de-iso-pentanized straight-run gasoline plus the normal pentanes from the platformate, will be blended into motor spirit. A sphere is being provided for the storage of iso-pentane. Heat for the process will be supplied by steam, low pressure for pre-heaters and high pressure for re-boilers. The de-iso-pentanizer operates at 25 S.P.I.G.

Works Visits

FROM September 1st, 1954, to May 31st, 1955, Wild-Barfield Electric Furnaces Ltd., Elecfurn Works, Watford By-pass, Watford, Herts, will be pleased to welcome parties of up to 30 senior engineering students from technical colleges and members of technical societies for tours of their Watford works. The visits are held on a number of week-day afternoons. Early application is advisable.

FORM GENERATION

A New B.S.A. Machine

A FORM generating machine with many interesting features has recently been developed by B.S.A. Tools Limited, Mackadown Lane, Birmingham, 33. By agreement with Pee-Wee Maschinen, this new machine is based on the well-known Pee-Wee No. 3 thread rolling generator, but it incorporates several modifications and refinements and is built entirely to British and American standards. The machine, without guards, is illustrated in Fig. 1.

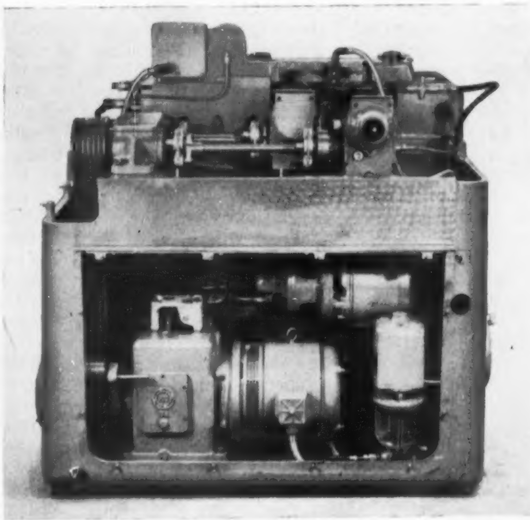


Fig. 1. Rear view of the B.S.A. form generator with covers removed

This B.S.A. generator is intended for universal application and is capable of rolling deep forms, such as Acme threads, worms and gears, see Fig. 2, in any material possessing suitable characteristics for cold forming. Multiple threads, providing the pitch is within the range of the machine, and tapered threads can also be produced on the standard model. Work between $\frac{3}{16}$ in and 3 in diameter can be accommodated and the machine will take rolls up to $6\frac{3}{4}$ in diameter and $6\frac{1}{2}$ in maximum width.

Of the special features incorporated in the design, perhaps the chief is the ease with which magazine and hopper feeding can be applied. By a completely new form of base construction, the usual top "stretcher" is eliminated and the space above the rolls left completely open. Any form of magazine attachment can therefore be arranged to feed directly between the rolls. Other special features are: provision for re-rolling on heavy components by means of spindle reversing control mechanism; an electronic timer that precisely controls the dwell period of the operating cycle; steplessly variable roll speeds by means of a handwheel-operated F.U. variable speed gear; and variable working pressure up to 15 tons maximum.

In thread or form rolling, two rotating, precision ground rolls are pressed into the surface of the cylindrical blank under hydraulic pressure. The component material becomes plastic under the pressure and is forced to flow into the thread, or other contour, of the rolls. Normally, the operation is carried out by the centreless method with the workpiece rotating freely between the rolls, on a rest and with its centre-line slightly below the centres of the forming rolls. Alternatively the component can be mounted between centres by the use of a special device.

Form generation has several advantages. In the first place it is easier to produce one pair of high precision form rolls to produce an accurate form on thousands of components than it is to obtain the same accuracy by machining the components individually. The dimensional accuracy obtained is at least equal to that of other methods and the speed of production is very high. Furthermore, rolled threads are generally stronger than those produced by cutting methods, since the material is caused to flow into its new form so that the grain flow follows the work contour. In fact, it is claimed that the tensile strength of a rolled thread is up to 30 per cent higher than that of cut or ground threads.

Another point of interest is that the rolls, or cylindrical dies, used in conjunction with the steplessly variable hydraulic feed on the B.S.A. machine, give, in effect, a die of infinite length. Therefore, the rate of rolling can be closely controlled to suit the characteristics of the material being worked. The scope of the machine is thereby extended to include a wide range of different materials, including many such as Monel-metal, Invar, Mazak, nickel-iron, copper and the Nimonics, which are normally considered unsuitable for thread rolling.

Normally the component is inserted between the rolls with no endwise motion and the length of thread is governed by the amount of blank inserted; the maximum length of thread by this method, plunge rolling, is equal to the width of the rolls. Work requiring extra long threads is rolled by the through-feed method whereby the blank is caused to pass between the rolls, and the length of thread is unlimited.

There are five main factors that determine the accuracy of this forming process: the blank diameter, the rolling pressure, the rolling time, the speed of the rolls and the rolling feed. These factors are, of course, inter-related, and on the B.S.A. form generator each is

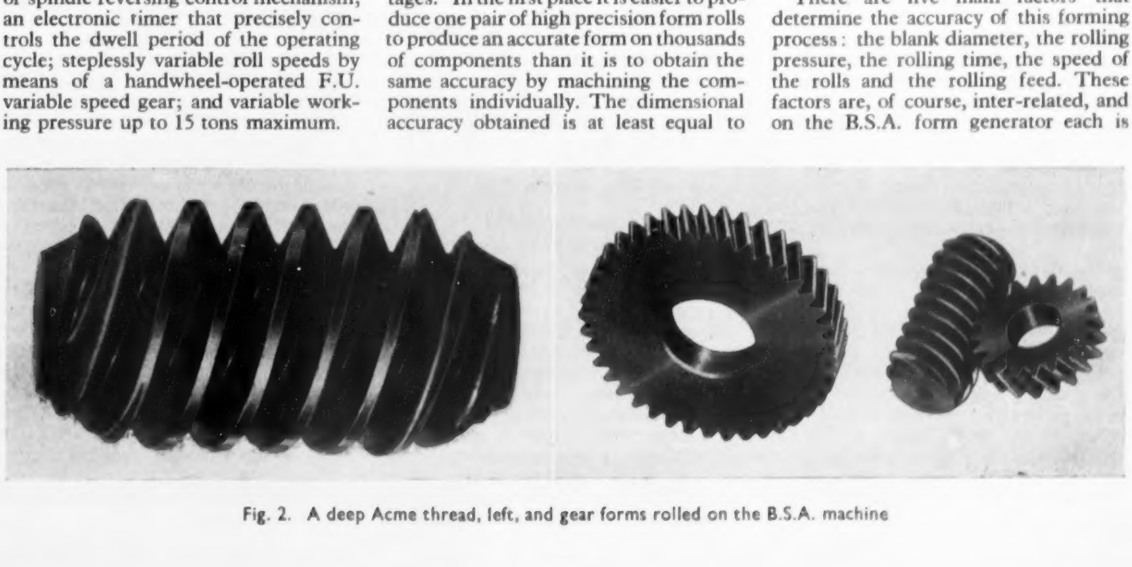


Fig. 2. A deep Acme thread, left, and gear forms rolled on the B.S.A. machine

widely adjustable so that there is great versatility of operation.

As was stated earlier, the machine lends itself to magazine or hopper feed. Where this method of feed can be adopted, one operator can control several machines. When very deep forms are to be produced, it may be necessary to roll the component more than once, and for this purpose a special re-roll valve has been incorporated in the design. This enables the spindles to be reversed automatically by the operation of a single lever. The spindle reversing control is such that the rolling pressure is automatically reduced, this is important to prevent distortion during sudden reversal, and the rolls are then reversed and the pressure re-applied. This action continues until the correct depth of form is reached, at which point the electronic timer trips out the hydraulic feed and the rolls are withdrawn.

A B.T.H. electronic timer is built into the machine. It automatically times the dwell period of the roll slide. In other words, only the final portion of the operating cycle is timed from a pre-set depth. This is to allow for variations in the work material, since the time taken to reach the pre-set depth at which the timer starts to function may vary from one piece of work to the next. By timing only the dwell or finishing portion of the work cycle, greater and more consistent accuracy is assured.

An F.U. variable speed gear couples the spindle driving motor to the gearbox. It provides steplessly variable control of the spindle speeds within the limits of the gearbox. All adjustments to the variable gear can be made through a handwheel on the front of the machine.

Improvements have been made to both the gearbox and the roll spindle mountings to eliminate backlash, a factor that can be most troublesome, particularly when fine serrations are being rolled. The two roll spindles are normally spring-loaded to give a degree of flexibility during rolling, thereby reducing the strain in the rolls and increasing their life. On the B.S.A. form generator a device is incorporated for positively locking the lateral movement when fine serrations are being rolled. An intermediate gear, which is adjustable into mesh with the driving and driven gears, has been built into the gearbox, to allow backlash to be taken up.

Forward movement of the right-hand roll slide is imparted by hydraulic pressure derived from a pressure pump driven by a separate electric motor. Return of the slide is effected by spring pressure. The valve mechanism controlling the hydraulic system allows for hand, semi-automatic and fully-automatic operation. For hand operation, both the forward and return movements of the slide are actuated manually and the timer is out of circuit. This method may be adopted for very intricate components for which individual control is advisable.

Semi-automatic operation merely involves initiating each cycle by hand: the timer automatically withdraws the rolls at the appropriate moment. Maximum output rates are, of course, obtained when fully-automatic operation is employed. With this method the cycle is completely automatic and is continuously repeated at a rate that depends solely upon the speed of loading and unloading. The working speed can be regulated to suit the loading speed.

The cast iron base cabinet of the machine is of very robust construction to support the machine bed firmly and eliminate vibration. All controls are situated on the right hand side at the front of the base casting. Two large oil tanks are housed in the base, one for coolant and the other for hydraulic oil. They are of such capacity that continuous operation is possible without any appreciable temperature rise. A gear pump circulates the coolant oil from the tank to the work area, whence it returns via a filter to the tank.

The left-hand roll spindle, the stationary member, is mounted in a reaction bracket bolted to the left-hand side of the table. This bracket can be swivelled round its pivot bolt so that the roll can be aligned with the right-hand thrust roll, or set at an angle for rolling taper threads. The right-hand roll spindle is mounted on a slide that is pushed forward by hydraulically-operated pistons. For normal centreless operation, the work rest carries a tungsten-carbide tipped blade to support the work. The blades are easily changed for work of different diameters or when replacement becomes necessary.

NYLON DROP STAMP BELTS

DROP stamp belting made from nylon is now being widely used in drop forges in this country. In some it has almost completely superseded hair belts and it seems that it will be only a matter of time before it is adopted universally. In drop forges where it is being used extensively it has been found that operating efficiency has increased, that costs have come down and that the loss of valuable production time has been considerably reduced. One belt, for example, which was fitted to a 4 cwt stamp, is still in use after four years. It is calculated that during that time it has made a total of 14,400,000 lifts and has raised a total weight of 2,800,000 tons.

Normally, a nylon belt can be expected to last about three times as long as a hair belt. As the nylon belt is frequently smaller than the hair belt it replaces, it is difficult to make any accurate comparison of the initial cost. On average, nylon is probably slightly more expensive. But this is more than offset by the much longer life of the nylon belt. In addition, production is increased and wasted time is reduced because the belts do not have to be replaced so frequently.

The economies are great. One of the biggest drop forge firms in the country, which first began to use nylon belts about two years ago, has so far not had to replace a single one because of normal wear and tear. On one 30 cwt hammer, the nine-inch, 14 mm, double hair belts which used to last between eight and 10 months were replaced with a single eight-inch, 10 mm, nylon belt in January 1953 and it is still in use. A 25 cwt dummy, which used to wear out a hair belt in six to eight months, has been using the same six-inch, 11 mm, nylon belt since August, 1952.

Head straps

Experience with head straps is even more impressive. Hair head straps used to last three or four days; nylon head straps are now lasting three or four months. When it is reckoned that the time lost in changing the head straps is between 20 or 30 minutes, the saving on this one item alone must run into hundreds of man-hours during the year in a large firm. There is also a considerable saving in the initial cost. The price of 2½ in, 8 mm, nylon head straps is little more than half that of the 2½ in, 11 mm, hair head straps which were formerly used. An additional advantage is that it is no longer necessary to carry such large quantities of spares, so useful storage space is freed for productive uses.

Drop stamp belting is among the severest of the 200 purposes for which nylon is being used at present. Yarn of a maximum tensile strength is used for weaving the belts and its elasticity (15 to 19 per cent extension at break) enables it to absorb the terrific shocks. Nylon's powers of shock absorption, incidentally, were graphically demonstrated during a test on a nylon rope. It was found that the nylon survived shocks that parted steel wire ropes of about three times the tensile strength. Another factor which has contributed to nylon's success as a material for drop stamp belts is its abrasion resistance which is greater than that of any other fibre. The fitting of the belts presents no complications. Metal clamping plates, belts or outboard clamps may be used. The latter are preferable as they avoid the necessity of having to make holes in the cloth. When holes must be made a pointed awl should be used to avoid damage which would be caused by a hollow punch.

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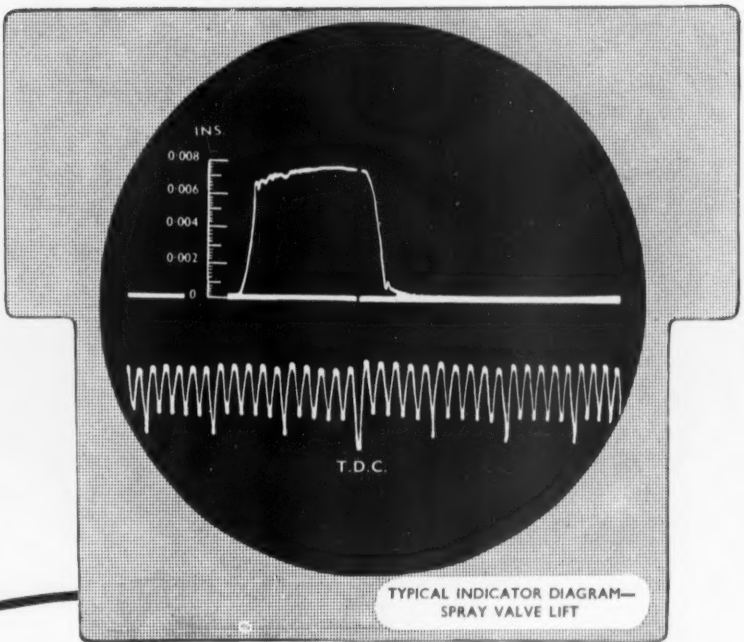
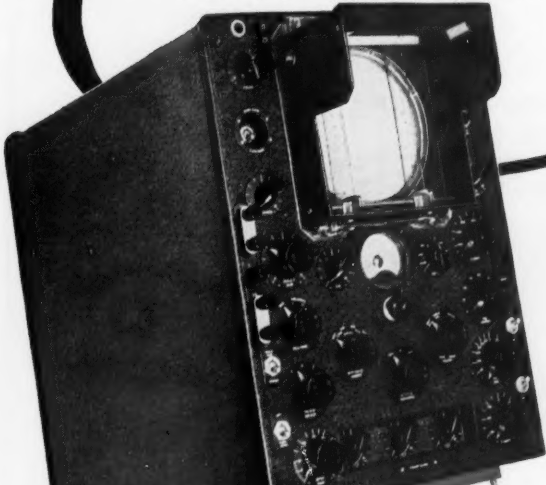


TYPICAL INDICATOR DIAGRAM—SPRAY VALVE LIFT

for

Engine Speeds: 80 to approximately 22 000 r.p.m. **Cylinder and Fuel Line Pressures:** up to approximately 10 000 lb/sq in. **Torsional Vibrations:** to approximately 60 000 c.p.m.



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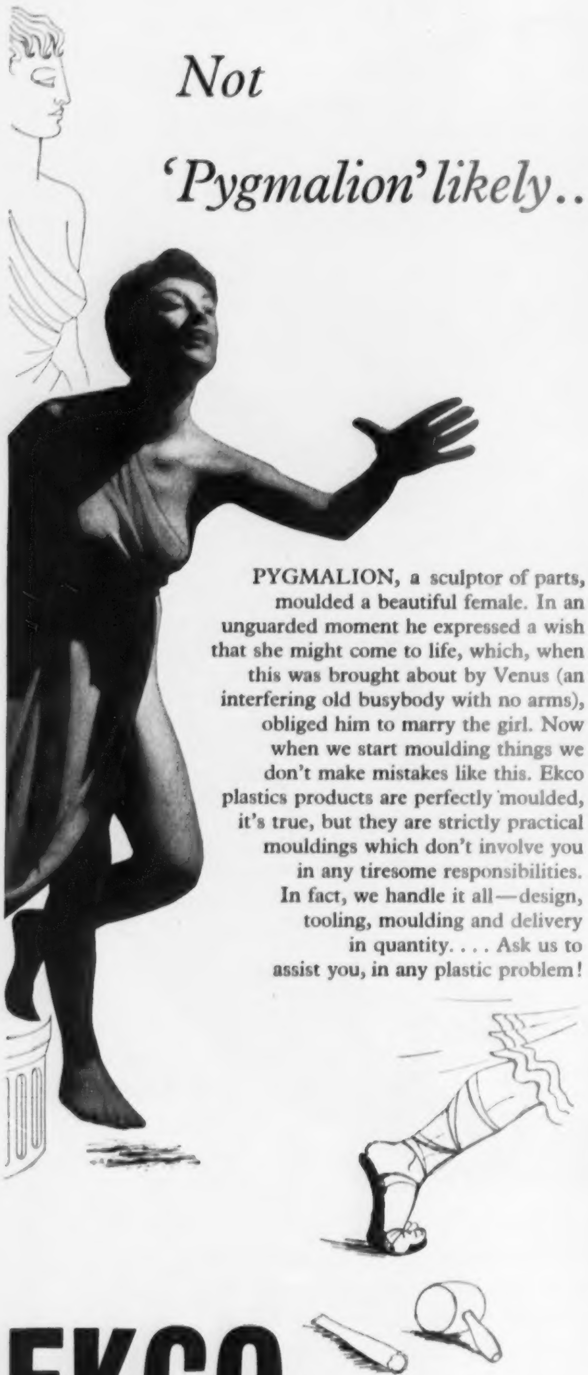

Not

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PYGMALION, a sculptor of parts, moulded a beautiful female. In an unguarded moment he expressed a wish that she might come to life, which, when this was brought about by Venus (an interfering old busybody with no arms), obliged him to marry the girl. Now when we start moulding things we don't make mistakes like this. Ekco plastics products are perfectly moulded, it's true, but they are strictly practical mouldings which don't involve you in any tiresome responsibilities. In fact, we handle it all—design, tooling, moulding and delivery in quantity. . . . Ask us to assist you, in any plastic problem!

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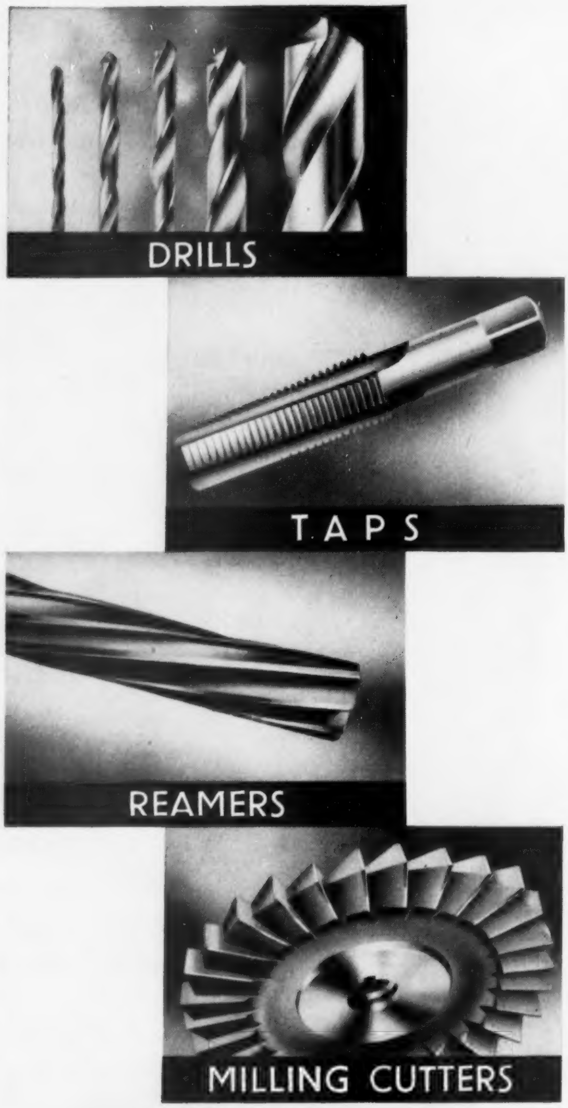

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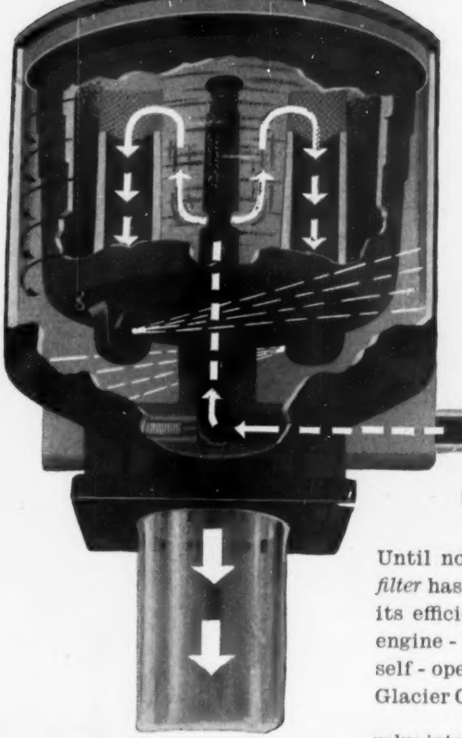

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FOOLPROOF EFFICIENCY—NO ELEMENTS TO REPLACE

Until now, to users of road vehicles and stationary engines, an oil *filter* has always meant an oil *strainer*. Centrifugal separation, though its efficiency was known, has never been generally available to the engine - building industry. Now it is here, in the form of a compact, self - operating component. And the advantages now offered by the Glacier Centrifugal Oil Filter are so many that they cannot be ignored.

HOW THE GLACIER CENTRIFUGAL WORKS

Like many another technical advance, the idea is basically simple. In place of filter-packs and elements, the Glacier harnesses centrifugal force (at a thousand times the force of gravity) to do the job better. Look at the sectional diagram, and you will see how *the oil itself* drives a rotor which flings the dirt to the sides of the bowl, while the clean oil flows unchecked through the hole in the base.

THE TEST OF EFFICIENCY

Too simple to work? The proof lies in working efficiency. The Glacier Centrifugal Oil Filter has been submitted to exhaustive tests and cannot be faulted. If oil is cleaned by passing through a filter medium on the principle of a strainer, efficiency at best is relative: *the Glacier Centrifugal intercepts particles that will pass through the mesh of the finest strainers in use.* Many strainers, moreover, progressively lose efficiency as they become clogged with the dirt they are trapping until the ever more restricted oil-flow and loss of pressure bring a by-pass valve into play which feeds dirty oil to the bearings. *The Glacier Centrifugal Oil Filter maintains high efficiency and constant flow.*

THE TEST OF £.s.d.

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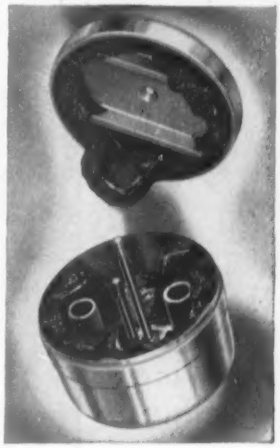

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1. Higher efficiency.
2. Constant flow.
3. No replacement elements.
4. No clogging.
5. Sump filtered every few minutes.
6. Dirt capacity many times greater.
7. Easy inspection and easy cleaning.
8. Easy fitting with Glacier accessories — no independent drive required.
9. A precision-engineering job which cannot go wrong.

***That's the* GLACIER CENTRIFUGAL OIL FILTER**

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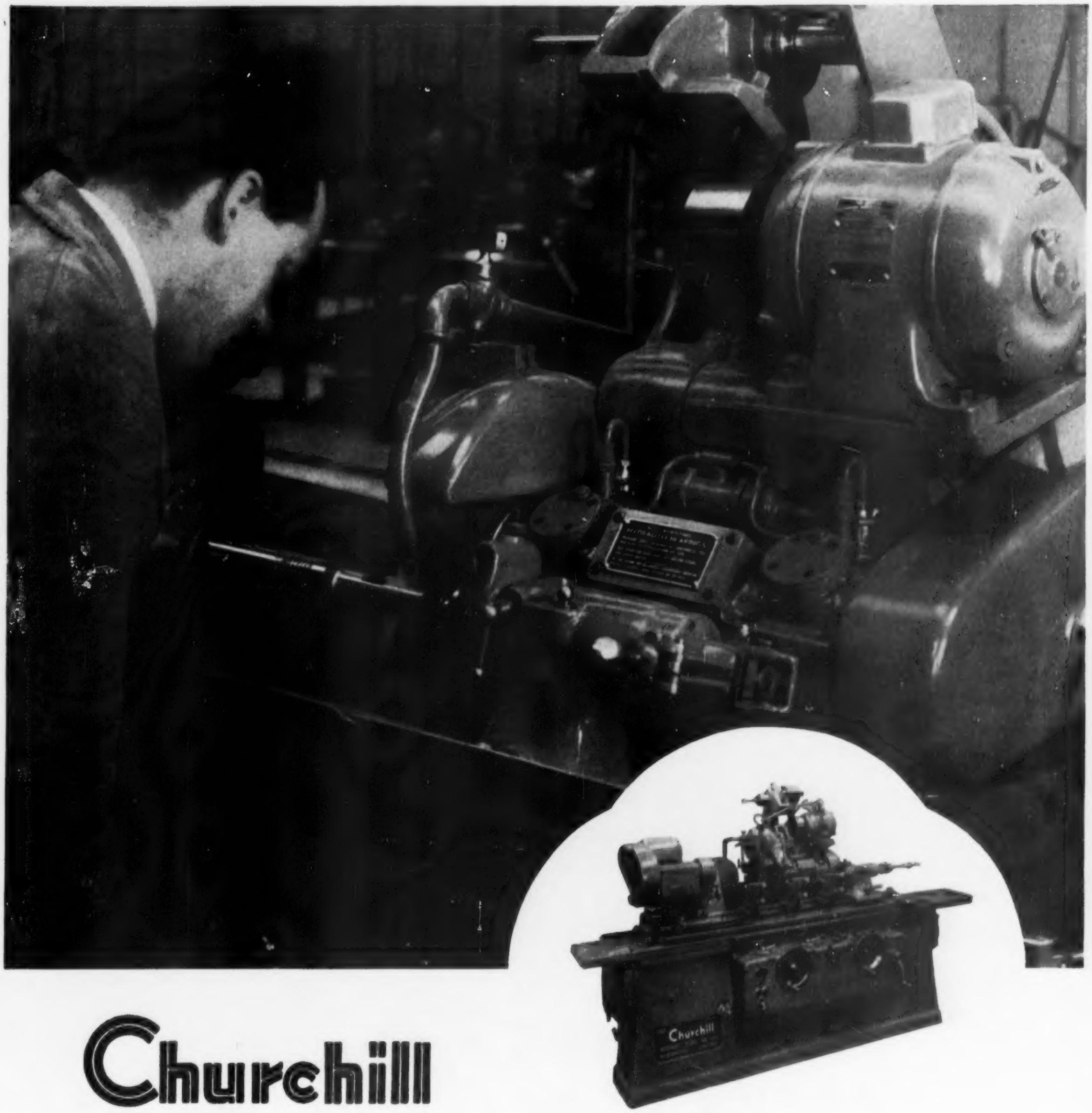

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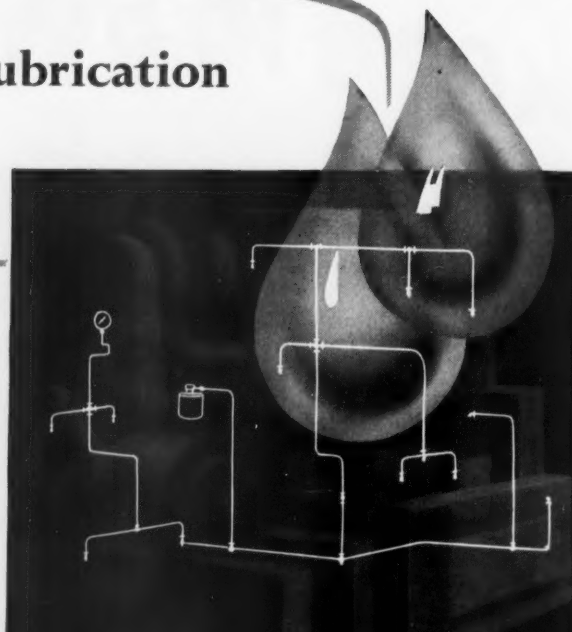

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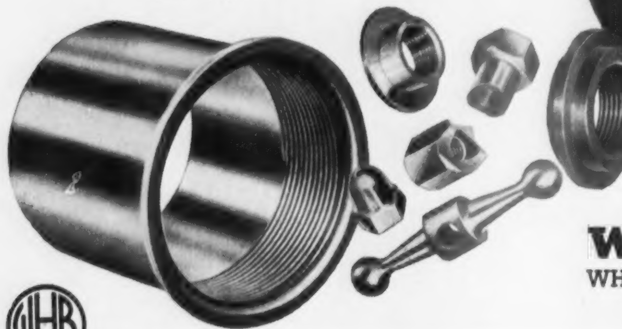

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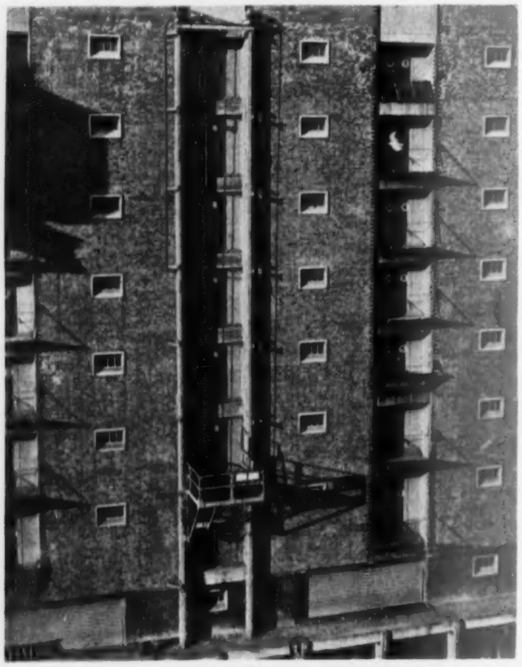

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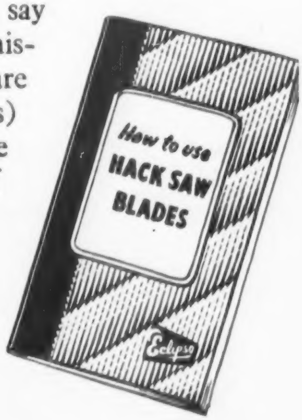

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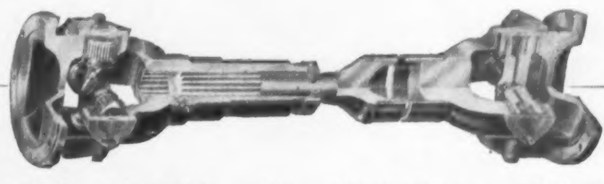

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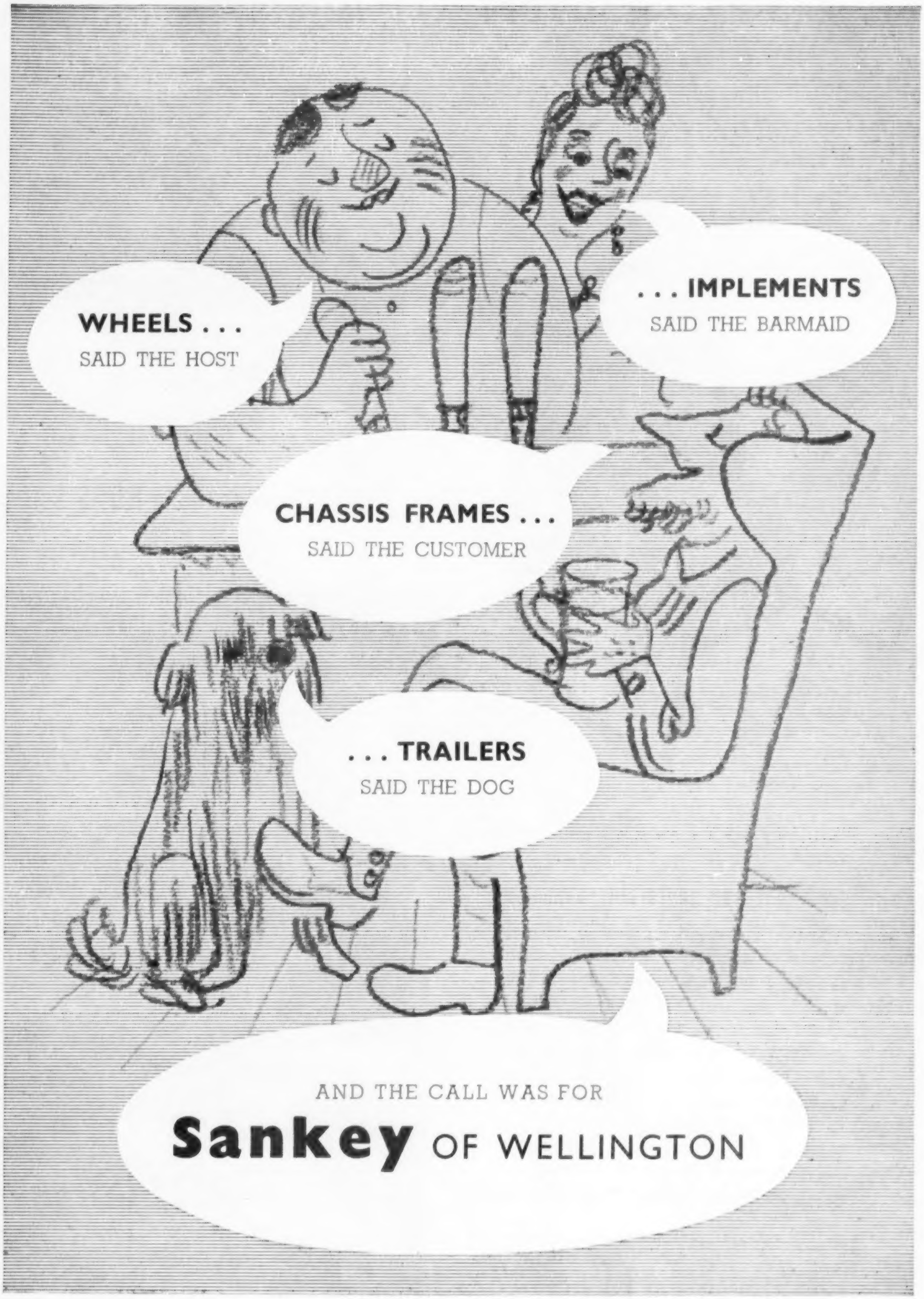
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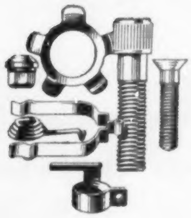

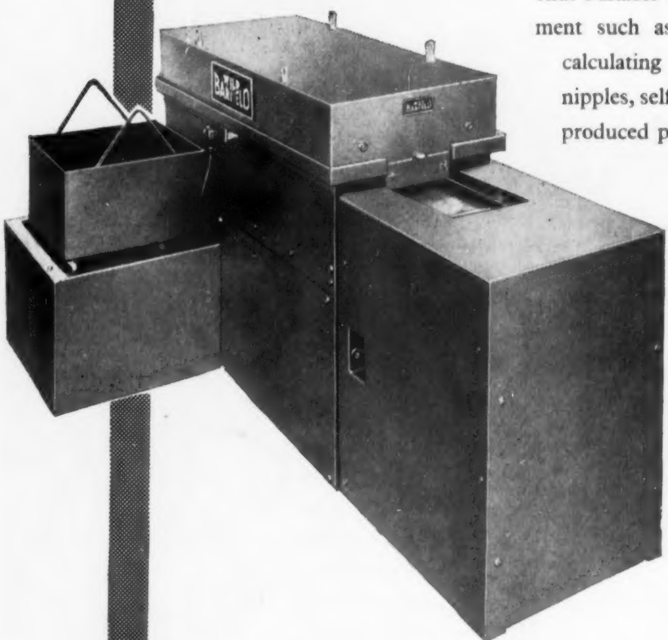

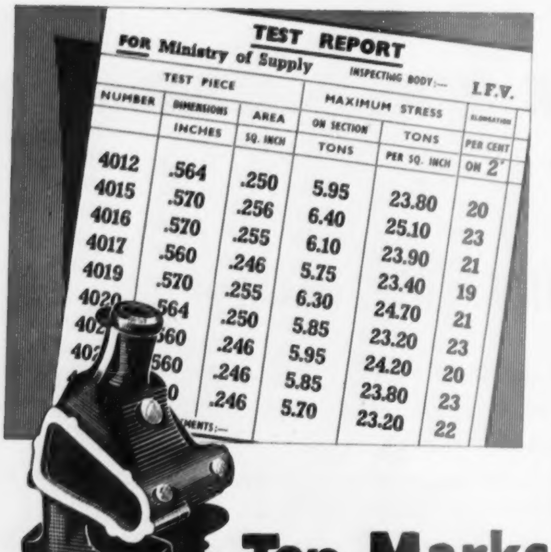

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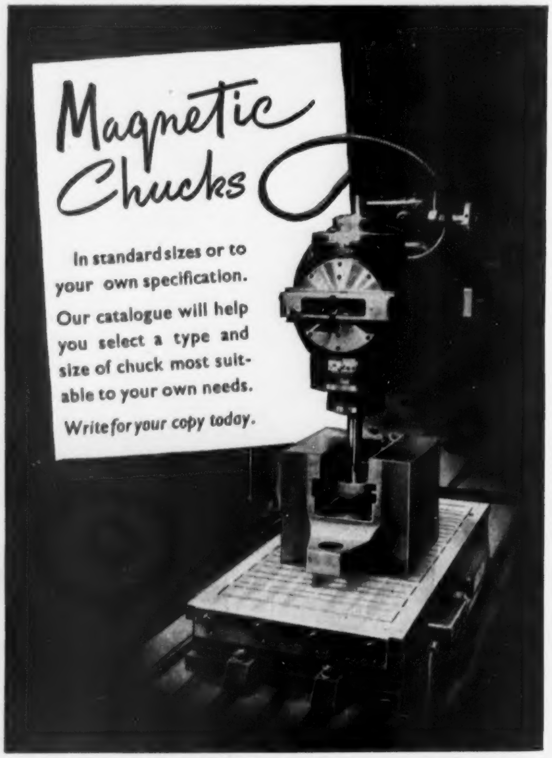

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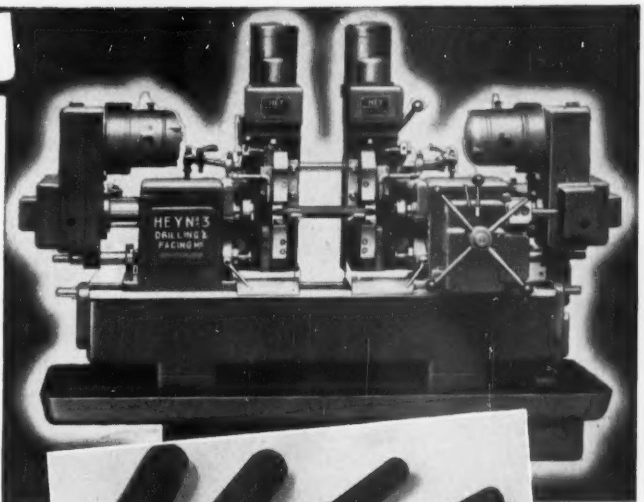

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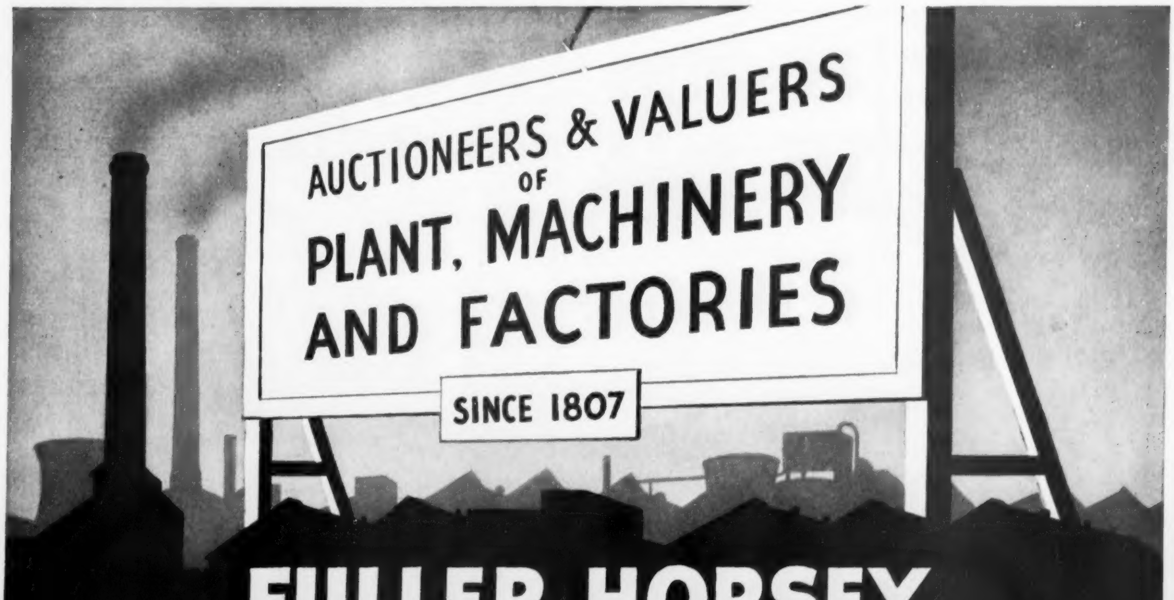

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AUTOMOBILE Service Engineers. Ford Motor Company Ltd. of Dagenham, Essex, require a number of men for expanding Service Department. Engineers with good all round automobile experience are required as instructors, technical assistants and claims correspondents. O.N.C. a desirable qualification. Permanent pensionable posts. Salary in accordance with experience and qualifications. Reply to Salaried Personnel Department, quoting reference RSP. [4838

SITUATIONS WANTED

EXPERIENCED representative (32) with entrée to design and purchase departments of all Midland automobile manufacturers, desires position with manufacturer requiring either established connection maintaining or development of existing connection. Box 6785 c/o Automobile Engineer. [4859

TENDERS

STOCKPORT CORPORATION TENDERS.

TENDERS are invited for Supply and Delivery to Lyme Park, Disley, of Four Passenger-Carrying Trailers. Drawings and specification from Manager, Lyme Hall, Disley, Nr. Stockport, and similar models can be seen at Lyme Park. Tenders to be sent to the Manager not later than first post on 23rd August. [4865

WANTED

AUTOMOBILE ENGINEER—Back numbers required: 1950, July; 1949, January, February; 1948, February, March, April, July, August, September, November, December. Offers to Box 6521 c/o Automobile Engineer. [4857

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